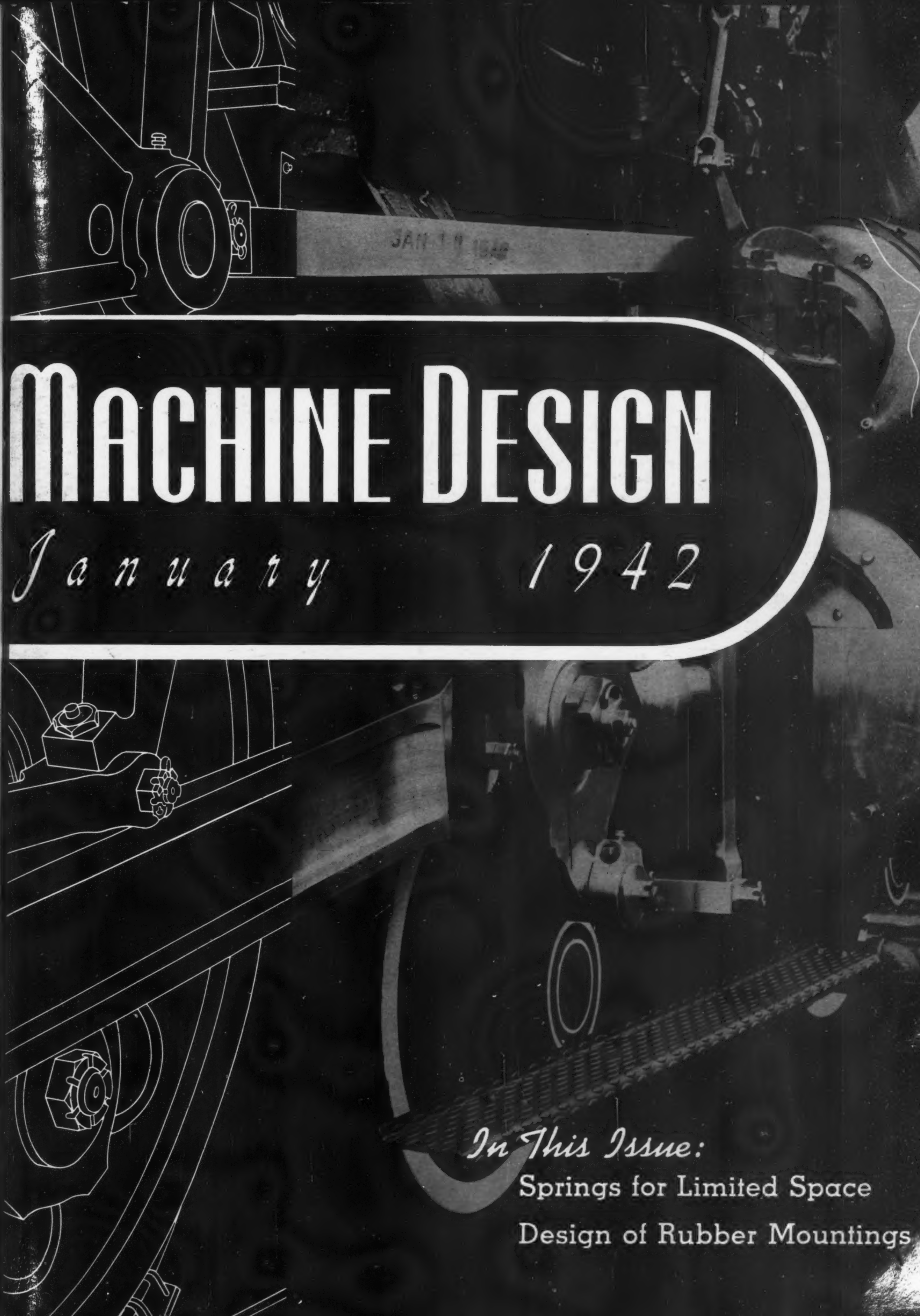


MACHINE DESIGN

January 1942

In This Issue:

Springs for Limited Space

Design of Rubber Mountings

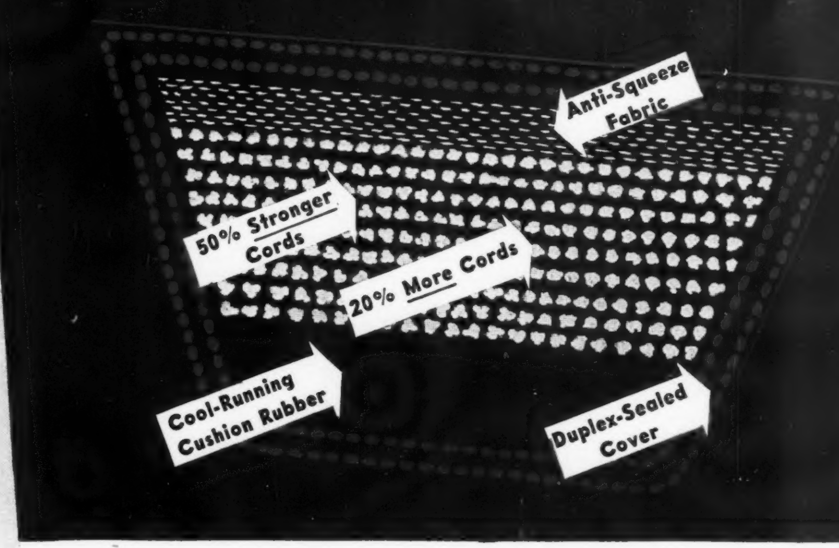
50% STRONGER CORDS
20% MORE CORDS
In Texrope* Super-7 V-belts!

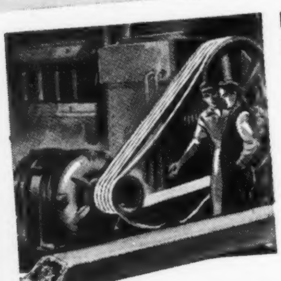
1 50% STRONGER CORDS! Sensational new Flexon process makes cords 50% stronger than even those in former great Texrope belts.

2 20% MORE CORDS! Count them! You get 20% more of these tough, extra pulling power cords in every Texrope Super-7.

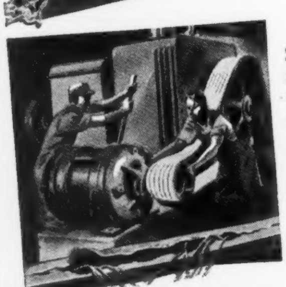
3 INCREASED LIFE! Cords float on amazing cool-running cushion rubber that absorbs shock . . . actually increases belt life phenomenally.

4 EXTRA PROTECTION! Exclusive duplex-sealed cover protects pulling cords against dirt, grit, moisture, and other adverse elements.





LOOK OUT FOR DANGER in belts with stiff, unpliable cords! Such belts *look* strong...but actually they buckle over sheaves, build up excessive heat that attacks pulling cords. Result: Belt failure far sooner than you expect it.



WARNING SIGN! Sagging, slipping belts mean too much stretch not backed up by balanced cord strength. Motor must be pulled back often to take up slack. Result: When stretch limit is reached belt breaks under tension.

SUPER-7 TEXROPE BELTS are made by amazing Flexon process that *combines* flexibility with low stretch...great strength. Cords float on cool-running, shock absorbing cushion rubber. Result: *true* strength . . . *true* pulling power . . . *true* endurance.

MORE production out of *present* equipment—that's the big need today. And that's what you get by changing over your power transmission to new Texrope Super-7 V-belt Drives!

For here's the newest V-belt line on the market—developed and built for modern high-speed production . . . to give you greater belt pulling strength and longer belt life . . . and add urgently needed efficiency to your machines.

You'll like these new belts — with 50% stronger "Flexonized" cords . . . 20% more cords . . . newly developed cool-running rubber cushion section to absorb operating shocks . . . "duplex-sealed" two-ply cover.

And you'll like their true V-shape for greatest groove-grip action . . . their silent, slipless operation.

So — do as hundreds of other hard-pressed companies are doing today. Use the new Super-7 V-belts to give increased production capacity to your present equipment.

Call your dealer or the Allis-Chalmers district office near you. Or write direct, asking for Bulletin B6190. A 1470

Available in All Sizes
½ hp to 2000 hp

*Trade mark registered U. S. Patent Office. Texrope Super-7 Belts are the result of the co-operative research and design genius of *two* great companies — Allis-Chalmers and B. F. Goodrich — and are sold exclusively by Allis-Chalmers.

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MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

Volume 14 JANUARY 1942 Number 1

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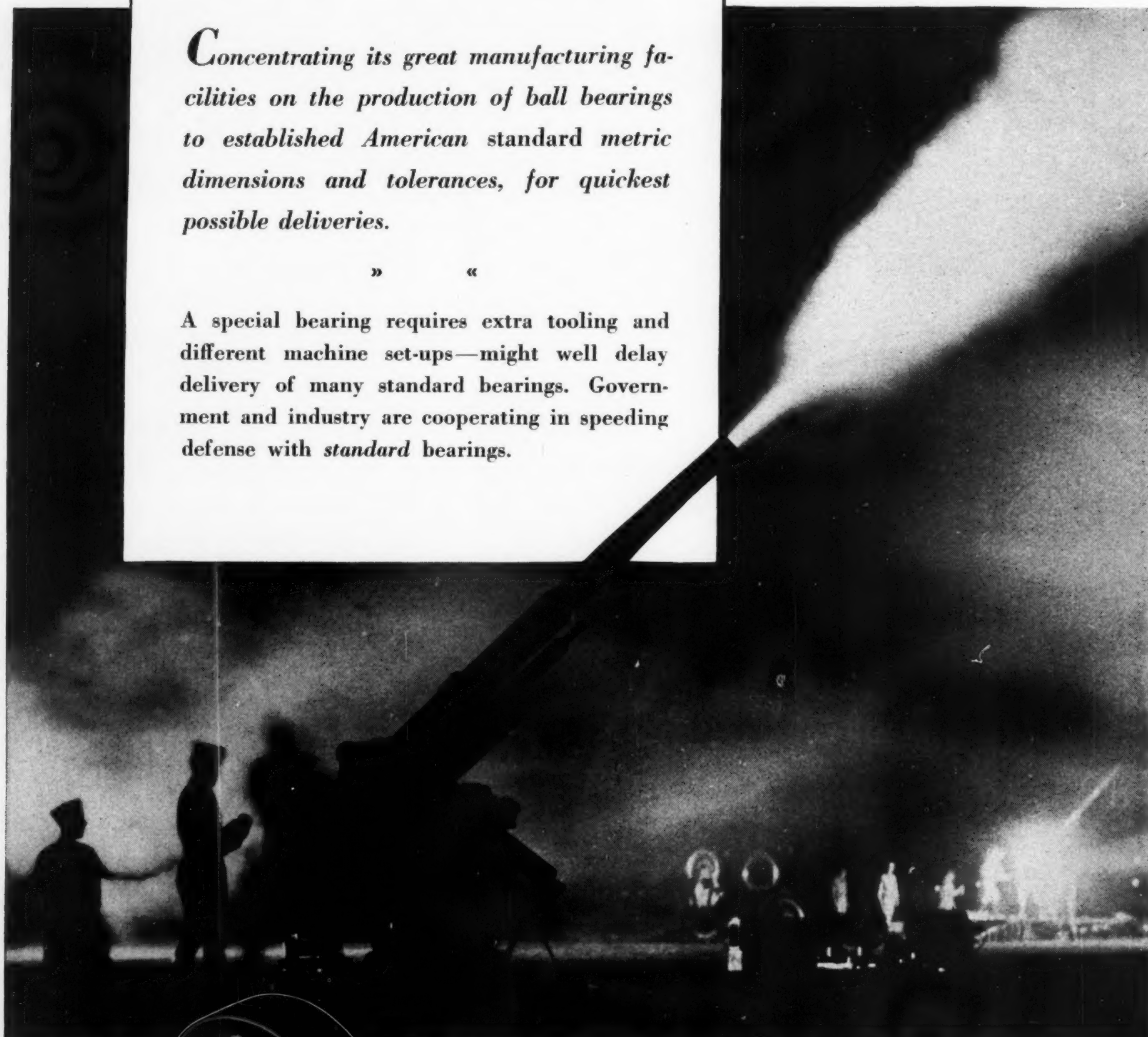
New Departure

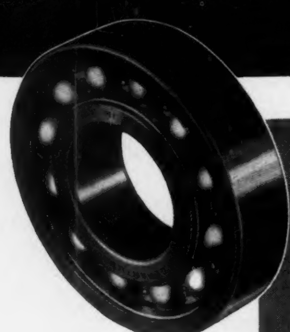
. . all out for defense

Concentrating its great manufacturing facilities on the production of ball bearings to established American standard *metric dimensions and tolerances, for quickest possible deliveries.*

» «

A special bearing requires extra tooling and different machine set-ups—might well delay delivery of many standard bearings. Government and industry are cooperating in speeding defense with *standard* bearings.





New Departure

THE FORGED STEEL BEARING

NOTHING ROLLS LIKE A BALL

3067

NEW DEPARTURE • DIVISION OF GENERAL MOTORS • BRISTOL, CONNECTICUT

Itemized Index

Classified for Convenience when Studying Specific Design Problems

MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.

AMERICA'S PIONEER PRODUCER OF MAGNESIUM REPORTS TO THE AMERICAN PEOPLE

EVERY AMERICAN is entitled to know how his own personal efforts in this great national emergency are being matched by fellow-workers in other defense activities.

We, therefore, consider it our duty to report on what we have done—are doing—and will do to meet the nation's mounting needs for magnesium.

Magnesium is the lightest of all structural metals—a full third lighter than aluminum. Its critical value is in contributing to American air supremacy—in saving those vital pounds that spell superior fighting speed—longer cruising range—greater load capacity.

Magnesium, therefore, is of deep interest to every American who holds dear the national heritages of our past—the hopes of our future.

Topics

IN RESPONSE to the oft-repeated question concerning the application of diesels to aircraft, Brigadier-General Kenney, Chief of Materiel division, Wright Field, says that little advantage could be gained from point of fuel consumption whereas greater hazard is suffered at high altitudes over present engine designs.

♦

AT AN all-time high, world nickel production is being increased by 50,000,000 pounds annually to take care of the demand arising from the war. During 1941, the United States consumed over two-thirds of the world's total production. Of the current production, steel mills are consuming 70 per cent, foundries and brass mills each 7 per cent, and special alloys and electroplating applications the remainder.

♦

CAPABLE of molding plastics into intricate shapes with one operation, a new Walco process is especially adapted to forming thermoplastic sheets into a wide variety of parts having compound curves and angles. This sheet-molded process eliminates many finishing operations and also makes practical the production of uniformly thin-walled units. Buffing or polishing is not required on edges or die seams.

♦

TYPICAL of the releasing of vital materials is the replacement of aluminum with steel in the "finger wheel," on dial telephones. This one item in itself released more than 135,000 pounds of aluminum during 1941. Other savings for telephone equipment include 300,000 pounds of nickel, 3,000,000 pounds of zinc, and 8,000 pounds of magnesium.

Further interesting examples of materials savings in various industries include: Replacement of stainless in thermostat units with silver, substitution for brass by using lead-antimony alloys in escutcheon plates, and silvered glass reflectors in lamp units to release brass and aluminum.

PREDICTING that within a year United States aircraft production would surpass the combined output of the rest of the world, Charles Dolan, president of the Lafayette Escadrille and retiring chairman of the aviation division of A. S. M. E., stated that production has already surpassed Germany's. Transformed within a year and a half into a three-billion dollar industry, none other has grown so fast in so short a time. It will soon surpass the all-time high of the automotive industry.

♦

IN THE new Ford aircraft engine plant, power for manufacturing operations is supplied by engines on test. Previously, engines wasted their power by driving test propellers. Now the units drive generators during both the shakedown or "green run" and final run tests, thus enabling them to feed power to the plant system. Hydraulic slip coupling between the engine and its generator is so designed that when the engine speed exceeds 720 revolutions per minute the generator runs at constant speed, delivering its power for manufacturing.

♦

ONLY one-year supply of rubber is on hand or afloat and headed this way, considering present annual consumption rate of 650,000 tons. Reductions in civilian use were a foregone conclusion. Because automobiles and trucks are vital to the war effort, suitable solution is imperative. Whether it be inferior tires, use of reclaimed rubber, synthetics, or controlled use, plans are in the making.

♦

RECENTLY approved, 42 new standards for aircraft engines have been developed by the Society of Automotive Engineers. Requested by the O. P. M., the standards include altitude graphs for engine performance; carburetor control connections; propeller shaft ends; tachometer drives; magnetos; bolts, screws and nuts; definitions; etc.

MACHINE DESIGN

Safe Speeds for Flywheels!

By Joseph Marin
Illinois Institute of Technology

ROTATING disks and cylinders are examples of machine parts in which a combined state of static stress occurs. As in the previous article on shafting[1], the current article deals with the design of these elements considering the distortion energy theory and in some cases the usual shear theory. In addition, design charts are given which afford a convenient means of easily selecting the allowable speed for a disk.

ROTATING SOLID CIRCULAR DISK: A circular disk may be distinguished from a cylinder by the fact that the thickness of the disk is small compared to its radius, whereas in a cylinder the length is several times the radius. For the solid circular disk shown in *Fig.* 1, at any distance r from the center there are two components of stress as shown in the figure. The values

[1] "Designing Shafting for Static or Fatigue Loads," MACHINE DESIGN, August, 1941.

INTEREST created among engineers by the author's article "Designing Shafting for Static or Fatigue Loads" in the August issue, as evidenced by numerous comments and requests for reprints, is indicative of the growing importance of stress calculations created by the war emergency. This article, and others by the same author on similar subjects to be published later, will be found equally valuable

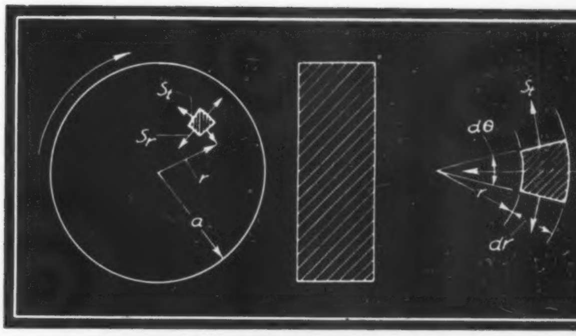
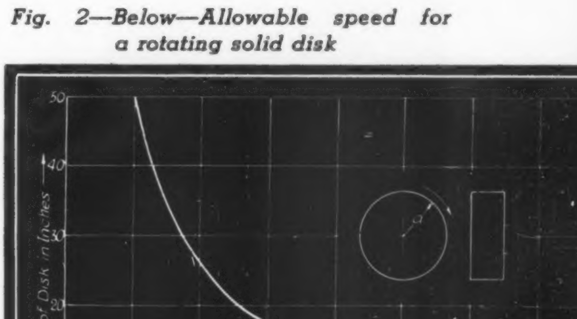


Fig. 1—Above—Components of stress for an element of a solid circular disk

Fig. 2—Below—Allowable speed for a rotating solid disk



of these stresses are[2]

$$S_r = k_1(a^2 - r^2),\ S_t = k_1(a^2 - k_2 r^2) \quad \text{(a)}$$

where $k_1 = \frac{1}{8}\,(3 + m)\,(\delta\omega^2)/g$, $k_2 = (1 + 3m)/(3 + m)$, in which m = Poisson's ratio, δ/g = mass per unit volume and ω = the angular velocity in radians per second.

Components Become Principal Stresses

Since there are no shearing stresses on the faces of the element shown in *Fig. 1b*, the components of stress are the principal stresses. Then the maximum shear stress within the element is

$$S_s = \left(\frac{S_t - S_r}{2}\right) = \frac{k_1 r^2}{2}(1 - k_2) = k_1 r^2\left(\frac{1-m}{3+m}\right)$$

or

$$S_s = \frac{S_r}{2} = \frac{k_1}{2}(a^2 - r^2) \quad \text{(b)}$$

or

$$S_s = \frac{S_t}{2} = \frac{k_1}{2}(a^2 - k_2 r^2)$$

Considering all possible values of r and m, the maximum value of the shear stress is, from these Equations,

$$(S_s)_{max} = \frac{k_1 a^2}{2} \quad \text{(c)}$$

By the *maximum shear theory*, the allowable speed is obtained by equating the value of the maximum shear stress in Equation c to the allowable value $S_w/2$, where S_w is the working stress in simple tension. This gives an allowable value for the magnitude of the speed

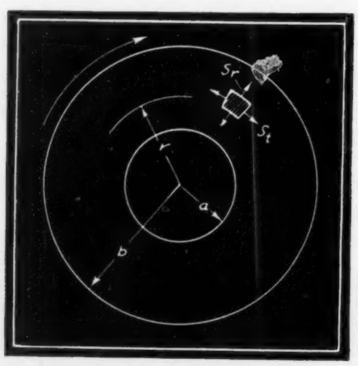


Fig. 3—Stress on an element of a hollow disk

$$\omega = \sqrt{\left(\frac{8g}{3+m}\right)\left(\frac{S_w}{\delta}\right)} \cdot \left(\frac{1}{a}\right) \quad \text{(1)}$$

Fig. 2 shows the variation in the allowable speed ω for different values of the disk diameter in inches.

[2] For a development of the stress analysis see, for example, S. Timoshenko, *Strength of Materials*, Vol. 2, Van Nostrand Book Co.

According to the *distortion energy theory* it will be necessary to consider the magnitude of the distortion energy for all possible values of r and to select the element for which this energy is a maximum. The expression for the distortion energy is

$$V = c(S_r^2 - S_r S_t + S_t^2) \quad \text{(d)}$$

where $c = (1 + m)/3E$.

Substituting the values of stress components

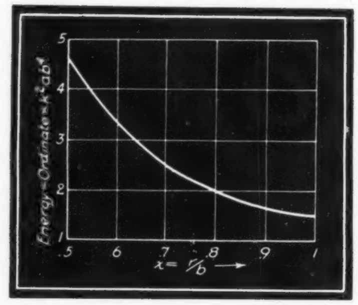


Fig. 4—Energy variation with respect to ratio r:b in hollow disk having inside to outside radius ratio of .5

from Equation a in Equation d, the distortion energy for an element at a distance r is

$$V = ck_1^2[(a^2 - r^2)^2 - (a^2 - r^2)(a^2 - k_2 r^2) + (a^2 - k_2 r^2)^2]$$

or

$$V = ck_1^2[a^4 - a^2(1 + k_2)r^2 + (k_2^2 - k_2 + 1)r^4] \quad \text{(e)}$$

Maximum or minimum value of the distortion energy occurs for a value of r defined by $dV/dr = 0$. From Equation e, this is for one of the following values of r

$$r = o, \text{ or } r = a\sqrt{\frac{1 + k_2}{2(k_2^2 - k_2 + 1)}} \quad \text{(f)}$$

Placing these values of r in Equation e, the maximum value of V is for $r = 0$, and its magnitude reduces to

$$V = ck_1^2 a^4 \quad \text{(g)}$$

For simple tension the allowable value of V is

$$V = cS_w^2 \quad \text{(h)}$$

Equating the values of the energy in Equations g and h gives the allowable speed as

$$\omega = \sqrt{\frac{8gS_w}{(3+m)\delta}} \cdot \frac{1}{a} \quad \text{(2)}$$

This equation is identical with Equation 1, as can be seen by noting that the two above theories coincide when the principal stresses at the critical element are tensile and equal in magnitude. For this particuluar case there is no difference between the

values as given by the older shear theory and the newer distortion energy theory.

ROTATING HOLLOW CIRCULAR DISK: For a hollow rotating disk, *Fig.* 3, the stresses at a distance r from the center of rotation are

$$S_r = k_1\left(b^2+a^2-r^2-\frac{a^2b^2}{r^2}\right)$$
$$S_t = k_1\left(b^2+a^2-k_2r^2+\frac{a^2b^2}{r^2}\right) \quad \text{(i)}$$

where the values of k_1 and k_2 are as in the case of the rotating solid circular disk.

Determining Allowable Speed

To determine an expression for the allowable speed in this case, only the distortion energy theory will be considered. The expression for the distortion energy in an element at a distance r is obtained by placing the values of the stresses from Equation *i* in Equation *d*. Doing this, the expression for V reduces to

$$V = k_1^2cb^4\left\{[1+(5-3k_2)\alpha^2+\alpha^4]-(1-\alpha^2)(1+k_2)x^2+(1-k_2+k_2^2)(x^4)+\frac{3\alpha^4}{x^4}\right\} \quad \text{(j)}$$

where $x = r/b$ and $\alpha = a/b$.

In order to determine the critical element or the point at which failure would occur according to the distortion energy theory, it is necessary to determine the maximum value of V. This can be done by using the condition for maximum V; namely, $dV/dx = 0$. Then the value of x for maximum V is defined by the equation

$$x^8-1.05(1-\alpha^2)x^6-4\alpha^4=0 \quad \text{(k)}$$

This equation does not give the value of x for maximum V as normally obtained so that it is necessary to consider the variation in V as x varies. A graph showing this variation is given in *Fig.* 4 for a value of $\alpha = a/b = .5$. This shows that the critical element is at the inner edge of the disk, that is, for a value $r = a$. At this point there is only a tangential stress so that the allowable speed can be obtained by placing the tangential stress S_t at $r = a$ equal to the allowable stress in tension. Then from Equation *i* the allowable speed is

$$\omega = \sqrt{\left(\frac{8gS_w}{(3+m)\delta}\right)\left(\frac{1}{2b^2+a^2-k_2a^2}\right)}$$

or

$$\omega = \sqrt{\frac{8g}{(3+m)\delta}\cdot\frac{S_w}{b^2}}\left(\sqrt{\frac{1}{2+\alpha^2-k_2\alpha^2}}\right) \quad \text{(3)}$$

Considering steel disks with Poisson's ratio $m = .3$, the allowable speed determined from Equation 2 is plotted in *Fig.* 5 for various values of the ratio $\alpha = a/b$.

ROTATING HOLLOW CIRCULAR DISK WITH BOUNDARY STRESSES: Sometimes there are radial pressures or tensile forces distributed uniformly along either the inner or outer perimeter of the disk, *Fig.* 6. Examples of such forces are radial pressure from a shaft upon which the disk is shrunk, or a centrifugal pull from parts attached to the rim. Under these conditions the resultant stresses can be obtained by superimposing the stresses due to the boundary forces on those stresses due to the inertia forces. This procedure leads to the following expressions for the stresses[2]

$$S_r = A+\frac{B}{r^2}-\beta_1\omega^2r^2 \quad \text{(l)}$$
$$S_t = A-\frac{B}{r^2}-\beta\omega^2r^2 \quad \text{(m)}$$

where $\beta_1 = (3+m)\delta/8g$, $\beta = (1+3m)\delta/8g$. In Equations *l* and *m*, the constants A and B are to be determined from the boundary conditions. Considering the case of a hollow circular disk shrunk on a shaft, the boundary conditions that the radial stress at $r = a$ is $-p$ and at $r = b$ is zero give the necessary conditions for determining A and B from Equation *l*. Using these conditions, the values of A and B become

$$A = \left(\frac{1}{b^2-a^2}\right)\left[\beta_1\omega^2(b^4-a^4)-a^2p\right] \quad \text{(n)}$$
$$B = (a^2b^2)\left(\frac{p}{a^2-b^2}-\beta\omega^2\right) \quad \text{(o)}$$

Stress value for a point in disk is now known in terms of the angular velocity and shrink-fit pressure. In order to determine the allowable speed it is necessary, as in the above problems, to find the position of the critical element. Using the distortion energy theory this element is the element for which the distortion energy is a maximum. The expression for the distortion energy for any element is ob-

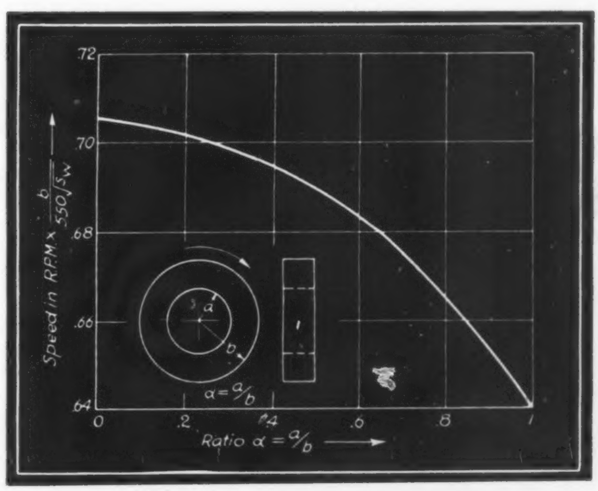


Fig. 5—Allowable speeds for rotating hollow disk

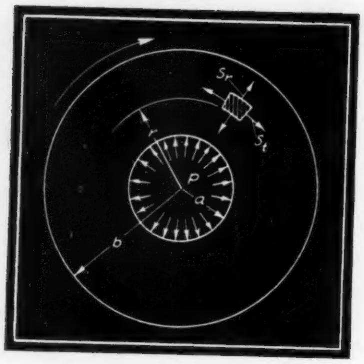


Fig. 6—Hollow disk with radial pressure caused by shrink fit on shaft

tained by placing the values of stresses from Equations l and m in Equation d. These substitutions give a value of the distortion energy which can be simplified to

$$V = c\beta_1^2 v^4 \left[\left(\frac{1 - k\alpha^2 - \alpha^4}{1 - \alpha^2} \right)^2 + \frac{(n-1)(k - \alpha^2 + 1)(3\alpha^2)}{(\alpha^2 - 1)} - x^2(n+1)\left(\frac{1 - \alpha^2 k - \alpha^4}{1 - \alpha^2} \right) + \frac{3\alpha^4}{x^4}\left(\frac{k^2 - \alpha^4 + 2\alpha^2 - 1}{\alpha^4 - 2\alpha^2 + 1} \right) + x^4(1 - n + n^2) \right] \quad \text{(p)}$$

where $v = \omega b$, $k = p/\beta_1 v^2$, $n = \beta/\beta_1 = (1 + 3m)$ $(3 + m)$, $x = r/b$, $\alpha = a/b$.

In order to determine the maximum value of V in this equation it is necessary to consider all possible values of x. Using calculus for defining the maximum value of V, an equation of the eighth degree is obtained. Under these circumstances it is more convenient to determine the variation in the value of V for all possible values of x and then to select the maximum value of V. An examination of Equation p shows that the value of x for maximum V will depend upon the magnitudes of p, v and α. For this reason it is desirable to substitute the numerical values of the constants in Equation p to determine the value of x for maximum V as illustrated by the following example:

Considering a hollow steel disk with $m = .3$, $\delta = .284$ pounds per cubic inch, $p = 5000$ pounds per square inch, $b = 10$ inches, $a = 5$ inches and $\omega = 1800$ revolutions per minute; then $\alpha = .5$, $k = p/\beta_1 v^2 = 8pg \div \delta(3 + m)v^2 = 4.67$, and $n = \beta/\beta_1 = (1 + 3m)/(3 + m) = .58$. Placing the values of these constants in Equation p, the distortion energy for an element at $r = bx$ is

$$V = 11{,}250c\left(2.37 + .49x^2 + .76x^4 + \frac{7.1}{x^4}\right) \quad \text{(q)}$$

The variation of V for different values of x is shown in *Fig.* 7. For this particular example the maximum value of V is for $x = .5$; that is, at the inner edge. To see whether this disk is safely designed it is only necessary to equate the energy in Equation q for $x = .5$ to the energy value in simple tension, which is $V = cS^2$. This shows that the equivalent simple tensile stress $S = 1150$ pounds per square inch. That is, the speed can be raised considerably over the value considered.

[3] For a discussion of the stress analysis see, for example, J. Prescott, *Applied Elasticity*.

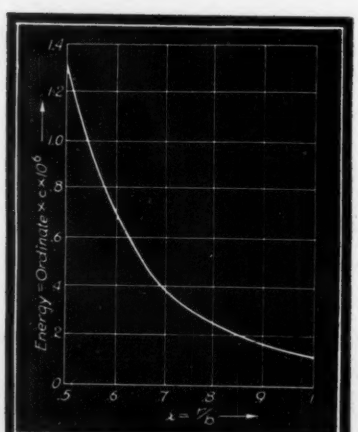


Fig. 7— Variation in distortion energy for disk with boundary stresses

Critical Element is at Inner Edge

If it is desired to obtain the maximum allowable speed for the above disk this can be done, assuming that the critical element is at the inner edge. Then equation p can be equated to the allowable value of $V = cS_w^2$ and with $x = .5$ the value of k, and thus the speed ω, is determined. With this value of ω it is possible by again using Equation p to find if the energy is maximum when $x = .5$.

Rotating Hollow Circular Cylinder: If the thickness of a hollow disk becomes large compared to its diameter we have a three-dimensional stress problem in which the three principal stresses, as shown in *Fig.* 8, are approximately[3]

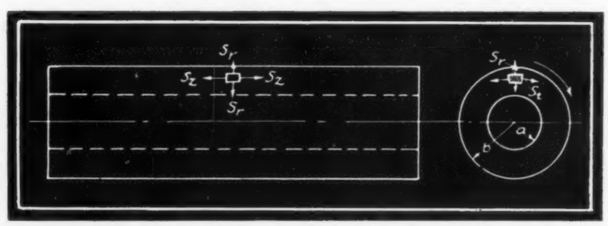


Fig. 8—Three-dimensional stresses on an element in a hollow cylinder

$$S_r = c_1\left(b^2 + a^2 - \frac{a^2 b^2}{r^2} - r^2\right) \quad \text{(r)}$$

$$S_t = c_1\left(b^2 + a^2 + \frac{a^2 b^2}{r^2} - c_3 r^2\right) \quad \text{(s)}$$

$$S_z = c_2(b^2 + a^2 - 2r^2) \quad \text{(t)}$$

where

$$c_1 = \frac{3 - 2m}{8(1 - m)}\left(\frac{\delta\omega^2}{g}\right)$$

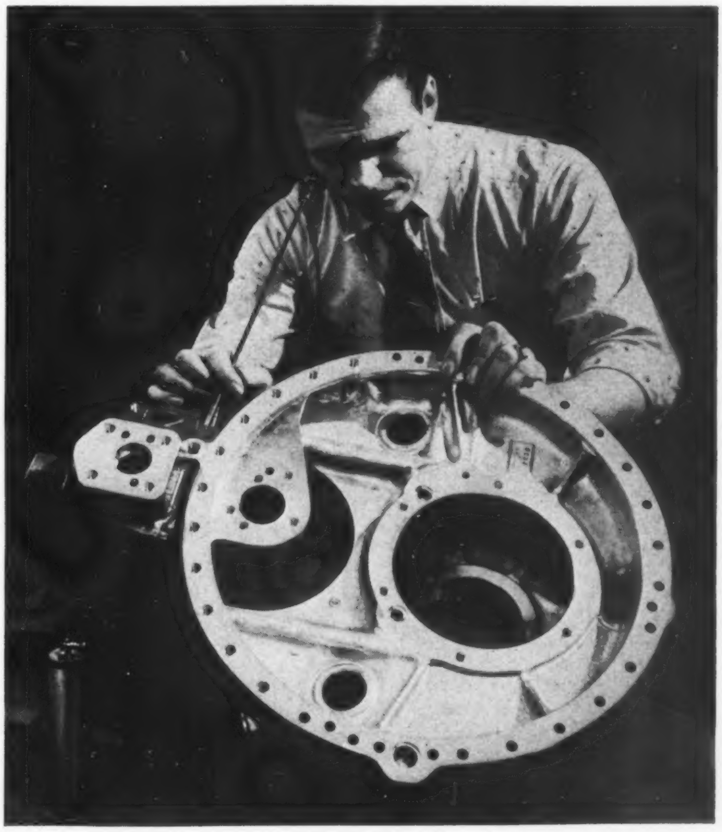
$$c_2 = \frac{m}{4(1 - m)}\left(\frac{\delta}{g}\,\omega^2\right), \quad c_3 = \frac{1 + 2m}{3 - 2m}$$

The above values for the stresses apply very well

(*Concluded on Page* 100)

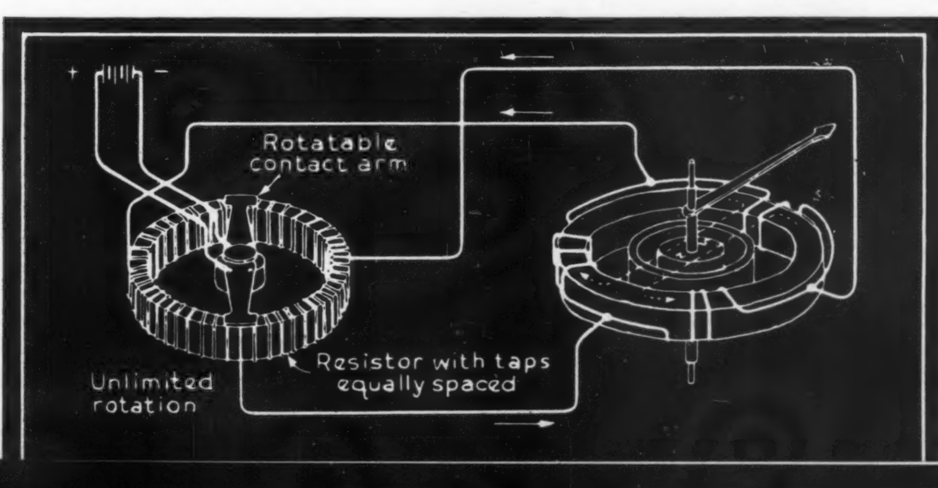
Scanning the Field for IDEAS

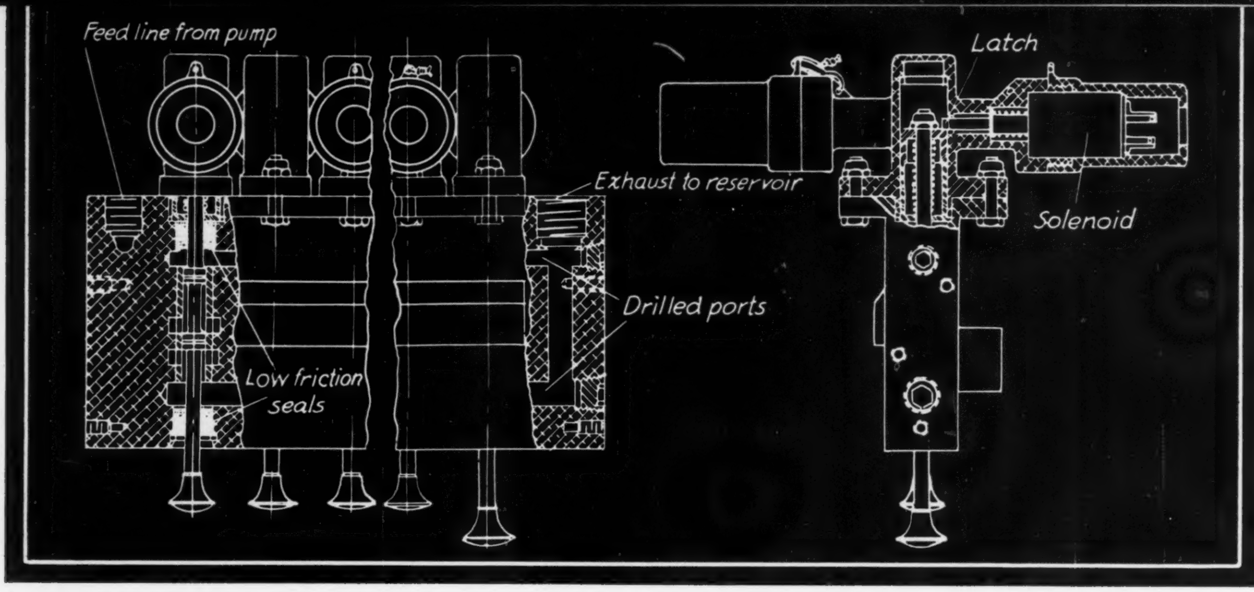
Visual inspection of tiny borings and fittings is aided by a borescope developed by General Motors. Resembling the surgeon's bronchoscope the instrument consists of .01-inch diameter light bulb together with a small mirror located in the tip. Cross reflections on the angled mirror and a microscope bring any burrs or uneven spots into relief. Illustration at right shows inspection of enclosed oil-line joints in a gear reduction case for an Allison aircraft engine.

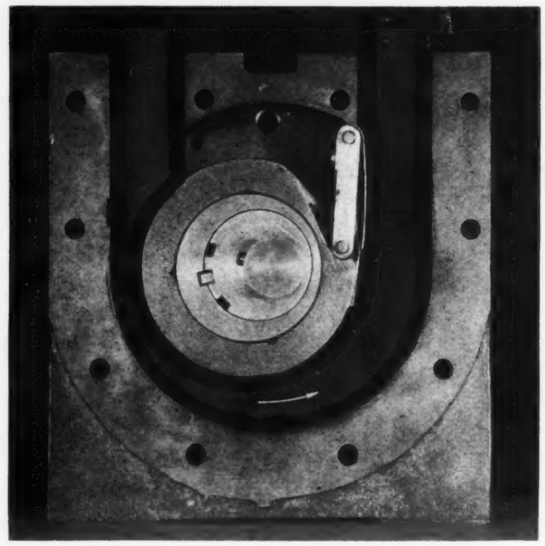


Counting ammunition rounds for guns in aircraft requires accurate, dependable and light-weight instruments for remote indication. Illustrated schematically below is the General Electric Selsyn method for indicating the number of rounds of ammunition remaining in each of the aircraft's machine guns.

Operating direct from the aircraft power supply, the transmitter consists of a tapped-resistance winding to which power is supplied through arms rotated by a shaft. The taps connect to three coils wound around a ring in the indicator. Distribution of voltage in the transmitter winding sets up a magnetic flux directly across the indicator ring, the position of which is dependent upon the voltage distribution in the three coils. Magnet on the indicator arm aligns itself with this flux, thus giving a duplicate position with the transmitter arm.







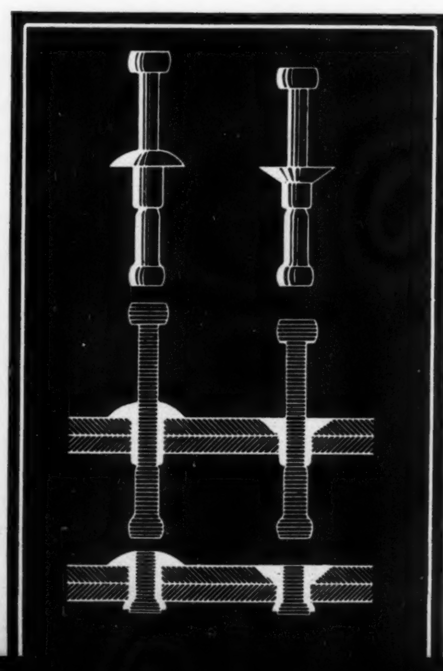
Integral design of hydraulic gun-charging valves for aircraft simplifies construction, reduces weight and saves space, as well as giving the gunner a more convenient control unit. In the drawing above is shown the control, designed by Fleetwings Inc., consisting of a mounting block which is drilled for port openings to obviate many fittings and joints.

To charge any combination of ten guns, corresponding valves are pulled. This action directs hydraulic fluid through the ports as shown to the gun charging mechanism. A solenoid release latch holds the valve open until the gun is loaded. When loaded, an electrical contact energizes the solenoid and valve is released for next charging operation. Previous designs utilized hydraulic releases with the possible disadvantage that, should a number of loading operations be performed simultaneously, surges in the lines might release a valve prematurely before loading.

Low-friction seals are important for dependable and easy operation in addition to precluding the possibility of seizing or scoring. A synthetic rubber, Ameripol, proved best for sealing against the low exhaust pressures. Operating pressure is 1500 lb. per sq. in.

Muscular action is simulated in the "squeegee" pump shown diagrammatically above by constricting a flexible tube and progressively advancing this constriction in the direction of pump action. Designed by the Huber Pump Division of Downingtom Mfg. Co., the U-shaped flexible tube is squeegeed by an adjustable impeller which has three keyed positions. These provide for differences in materials being pumped, whether dense or light liquids or gas. To prevent impeller action from stretching the tube on intake side of pump, the impeller is anchored by a link, giving a rocking action to the compression ring around the adjustable eccentric.

Blind rivets for inaccessible places have taken many forms. Usually, a designer carefully plans to avoid their use, but in many types of construction this is not always practical. A new rivet, illustrated at right, designed by Cherry Rivet Co., saves considerable design and assembly time and yields a dependable joint. This self-plugging rivet has a mandrel with an expanded section and head on the blind side. When pulled into the hollow member by a hydraulic tool the expanded section enlarges the shank and forms a tulip head in the back. Force required to apply the rivet breaks the mandrel. Strength and fatigue compare favorably with that of conventional solid rivets.



Basic Do's and Don't's in Applying Plastic Moldings

By W. B. Hoey
Bakelite Corp.

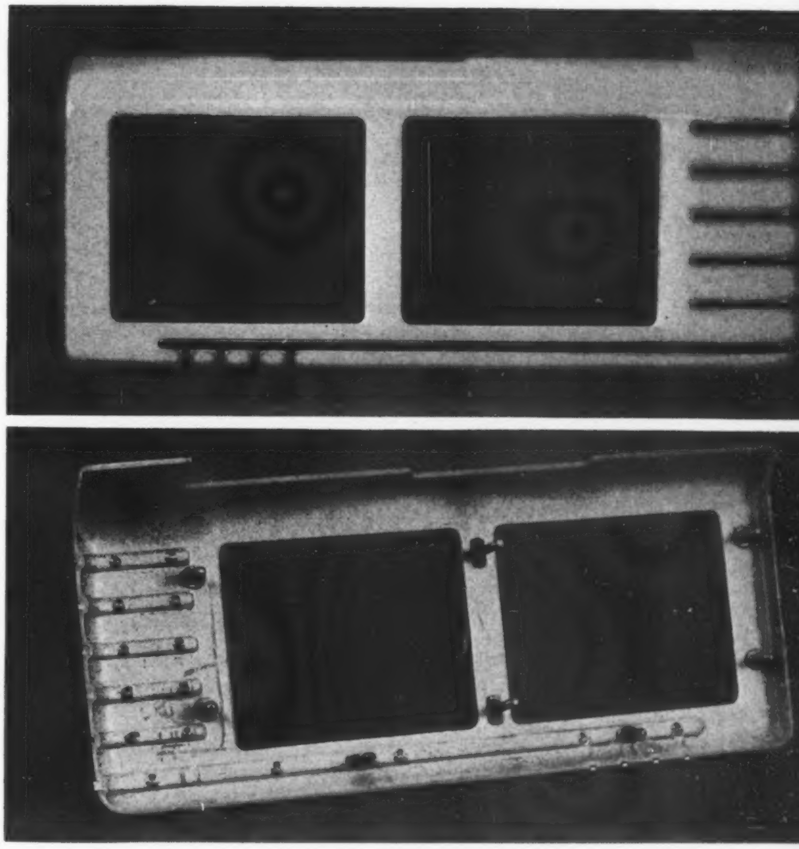


Fig. 1—Automotive instrument panel would be more satisfactory if molded of thermosetting urea instead of using metal stamping as insert with injection-molded plastic covering it. Bottom view shows back

WHEN an engineer designs a molded part he often thinks in terms of more familiar materials rather than in terms of plastics. He selects a plastic because he wishes to incorporate into his design some property such as inbred color, lightness of weight, heat or chemical resistance, electric insulation, or because he can eliminate costly assembly operations. He overlooks the fact that plastics have their own peculiar limitations and that there are also limitations in the fabrication of the molded part.

Considerable information and advice on the design of molded plastic parts has been published and it is unnecessary to repeat these instructions again here. D. M. Buchanan ably summarized* this very essential information. To render a part practical from the standpoint of economical production, such features as uniform wall thickness, side wall tapers, proper use of fillets, threads, metal inserts, the elimination of side draws, undercuts, etc., must be considered.

* *"Design Problems of Molded Plastic Parts"*—D. M. Buchanan, *Mechanical Engineering*, December, 1937.

THIS fundamental discussion, abstracted from a recent A.S.M.E. paper, will be useful to the engineer who is giving initial consideration to the application of molded plastics in machines. The author stresses the importance of understanding molding technique as well as the characteristics of the materials themselves. By citing examples he illustrates some of the pitfalls that may be encountered and their correction

There are several types of plastics, and the right one must be selected for a given job if it is to give satisfactory performance. For molding there are two general methods, and some types of materials can be fabricated by either method while others are limited to one. Both methods—compression and injection—are illustrated graphically in *Fig.* 2. Several types of molds can be employed and in many instances the design of the mold has much to do with satisfactory performance of the molded part in service.

It may be felt by some that mold design is a problem for the tool engineer and the production department, but the designers who obtain the best results are familiar with mold types and their construction. In many instances, bad features of a design can be eliminated by recognizing the diffi-

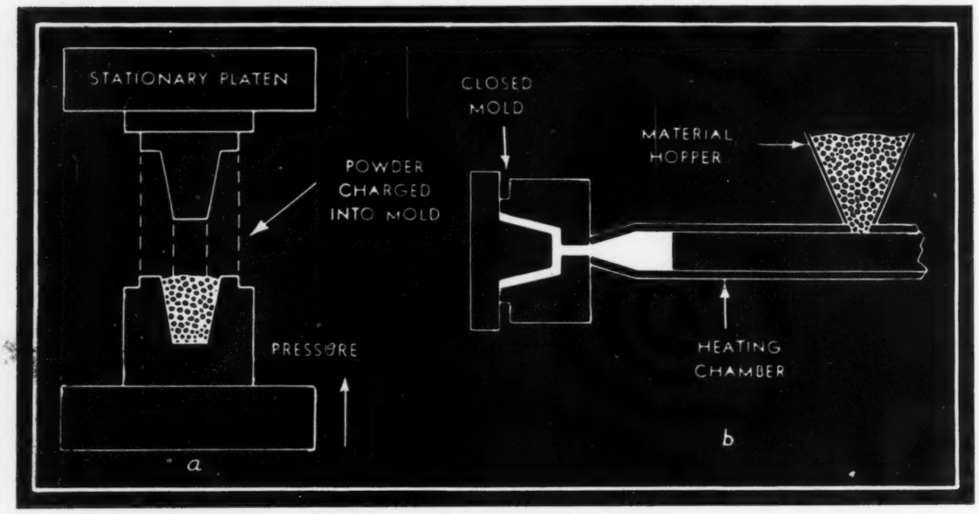
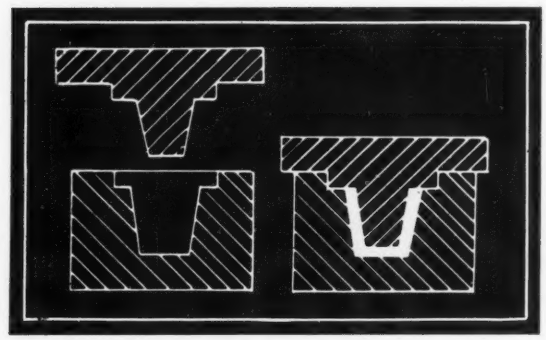


Fig. 2—Left—Basic principles of compression and injection molding shown at (a) and (b), respectively

Fig. 3—Below—Flash mold is simplest type of construction mold. It requires excess material for flash



culties which might be encountered in laying out and building the mold.

In compression molding, the material charge, either granular powder or more often a preformed pellet, is loaded directly into the mold cavity. Pressure is applied on the plunger or force plug and, as the mold closes, the material becomes sufficiently plastic to flow into the various mold crevices. Injection molding is that process by which the granular molding material is fed continuously from a hopper into a heating chamber. The material is softened into a plastic state in the chamber and then forced through an orifice into a closed mold. This method is faster than compression molding and for this reason smaller capacity (fewer cavities) molds can be used to obtain equivalent production of parts.

Both thermosetting and thermoplastic materials can be molded by compression, but this method is used chiefly for the former class. The latter are usually fabricated by the injection process. Until recently it has not been possible to mold the thermosetting materials by the continuous injection method because they would polymerize in the heating chamber. The thermoplastic polystyrene can be molded by compression, but the physical properties are improved if injection molding is used.

Types of Molds

Of the several kinds of molds used in compression work, they can all be considered as variations of three general types which are referred to as

1. Flash molds
2. Positive molds
3. Semipositive molds.

The flash or "overflow" mold shown in *Fig.* 3 is the oldest type and was borrowed from the rubber industry. It is the easiest to design and build. In its simplest form it consists of a top plate or "force" and a bottom plate or cavity.

In compression molding, the molding material (either powder or preformed tablets) is placed into the cavity. An excess of material must be used and, as the dies close, this excess is squeezed out into the overflow or sprue groove and forms a thin flash. A land area provides a positive stop of the mold halves and permits reasonably good dimensional tolerances in the direction in which pressure is applied. The flash type of mold is used in many molding plants for producing small articles.

Principal difference between the flash mold and a full positive mold, *Fig.* 4, is that there is no stop for the force plate in the latter. The full pressure is exerted upon the material in the cavity and the height of the molded article is determined by the amount of material used and the pressure exerted upon it. Full positive molds are limited to the compression molding of large parts when bulky materials such as macerated-fabric-filled phenolics are used. When multiple-cavity molds are necessary, the positive type of mold is slow to handle because of the necessity of charging the several cavities with exactly the same amounts of molding material.

Semipositive molds *Fig.* 5, combine the best features of the flash and positive molds. The upper section telescopes into the cavity, providing a seal against the escapement of an excessive amount of material. It provides also for a positive stop of the mold halves. Because of the telescoping action, only a small amount of material escapes.

This results in a decidedly greater density of the molded part adjacent to the mold parting line, closer tolerances on the height of the part and a saving of material. For these reasons, the semipositive mold is to be recommended for the compression molding of all types of materials, with but few exceptions, to insure the highest quality.

No Flash from Injection Mold

Injection molds are almost universally similar in type to the compression flash mold. However, in injection molding there is no flashing or overflow of excess material because the mold is already held in a closed position by high pressure before the plastic material is forced into it from the nozzle of the plasticizing cylinder through the sprue hole.

To insure maximum quality in the molded part, the size and location of gates and runners must be given careful consideration. The runners are

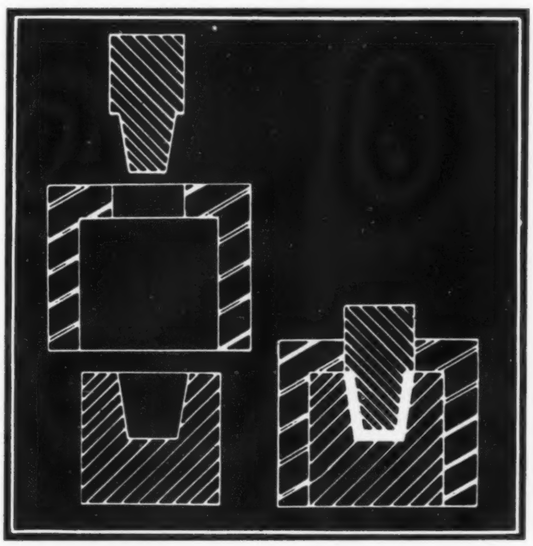


Fig. 4—Positive molds require three parts and are limited to compression molding of large pieces using bulky materials

the channels through which the plastic material passes from the sprue hole to the individual cavities of the mold. The gate is the restricted end of the runner where the latter enters the mold.

When runners and gates are held to a minimum size, the amount of sprue scrap is reduced. On the other hand, too much restriction of the flow of material results in lack of density, causing "flow lines", "sink marks" and possible stresses and strains which may lower the inherent strength of the molded part.

No definite rule can be given for determining the size of the runner and gate because there are such factors involved as shape, size and thickness of the molded part, the pressure available in the machine and the characteristics of the particular material from which the part is to be molded.

The location of the gating is also important because of its influence on proper welding. When the molding material is injected into the closed mold, it spreads out from the gate in all possible directions in the mold cavity and doubles back and welds upon itself to fill out the cavity. Welds are weak points in the part and the gating should be so located that this weakness will have a minimum effect on the performance of the product. In general the gates should be located as near as possible to the outside surface of the part. If there is variation in the section of the part, the gate should be at the thicker portion if possible.

Perhaps the results of failing to consider the limitations of plastic materials and molding methods can be emphasized best by citing some applications which did not give satisfactory performance in service.

Molded Cover Proved Unsatisfactory

Three years ago one of the automotive manufacturers decided to use plastics for an instrument panel and glove compartment door. Since the main purpose was to achieve unusual decorative effects, cellulose acetate was selected because of its color possibilities. A thin film of acetate was injection molded over a steel stamping, *Fig.* 1 The plastic gave the desired decorative effect and the stamping provided rigidity and strength required to withstand service abuse.

Flowing such a thin film of acetate over the relatively large area, however, proved to be a difficult production problem. It was found necessary to use a cellulose acetate formulation containing an unusually large amount of plasticizer. The plasticizer was a volatile and, in gradually escaping, caused age shrinkage with consequent cracking and splitting over the surface. In addition, the plasticizer attacked the paint on the metal cowling and caused it to lift and peel.

The problem was solved by substituting acetobutyrate. But, in this particular case, there is a question as to whether any plastic should have been used at all. Since it was necessary to use a

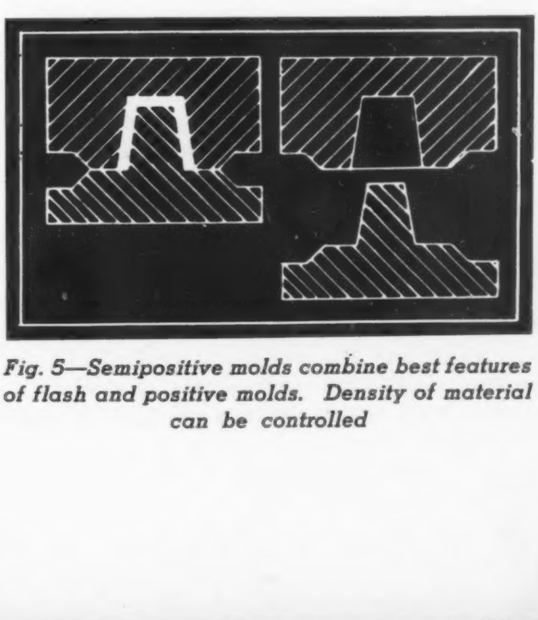


Fig. 5—Semipositive molds combine best features of flash and positive molds. Density of material can be controlled

steel stamping insert over the whole area, it would seem that an equally desirable decorative effect could have been obtained at lower cost by simply painting the stamping. On the other hand, the stamping could probably have been eliminated by using the thermosetting urea instead of cellulose acetate, but the changes in design necessary for the elimination of the metal insert would have required design changes in other parts of the cowling which were impractical at the time.

Limitations Were Overlooked

Errors caused by lack of knowledge of materials limitations and fabricating methods are also typified by the automotive radio aerial insulator shown in *Fig.* 6. Styling was considered paramount and the limitations of materials and fabrication were given little or no consideration. As a result, the product was a complete failure in service. A flock-filled phenolic material was chosen because of the possible shock which might be encountered in service. Being a peculiar shape the mold was complicated, and it was difficult to fill the tip completely. This meant that there was structural weakness in the area. To aggravate this weakness further, the body was molded solid and the ¼-inch hole for the metal aerial rod was drilled afterward. The part would have been considerably stronger if the hole had been molded rather than drilled. Also, the designer failed to take into consideration the fact that the insulator would be subjected to more severe flexural strains than shock because of the constant whipping of the aerial when the car was in motion.

Flexural Strains Control Design

Shown in *Fig.* 7 is a fuse holder cover which is an example of inadequate study of service conditions and molding requirements. The cover is hinged at the top and must be able to withstand the shock of slamming shut while carrying a heavy fuse. The first covers utilized a phenolic fabric-filled material because it was assumed that the main requirement was maximum shock resistance to withstand the slamming.

Breakage was encountered just below the ring handle. A study revealed that the slamming shut developed a flexural strain and the impact load was not as severe as had been anticipated. Also, the material did not form a dense molding in the areas near the boss and handle because of its inability to fill out properly around the obstructions in the mold cavity.

Changing to a general purpose, wood-flour filled phenolic eliminated the trouble. The impact strength of this material was only one-tenth that of the fabric-filled product, according to A.S.T.M. test ratings, but the actual difference in this part was considerably less because the general purpose material filled the cavity and produced a molding of proper density and normal strength. Furthermore, the flexural strength of the general purpose material was about 25 per cent higher than that of the fabric-filled material on the A.S.T.M. ratings and the actual difference on this particular part was even higher, due again to the inability of the fabric-filled material to mold into a sound part.

To illustrate the importance of the semipositive mold in compression molding, two examples will be discussed. An ignition coil case, approximately the size of a drinking tumbler and with a flange around the open end, was started into production in a flash-type mold of twelve cavities. The material used was a wood-filled, general-purpose phenolic. The case was assembled to a base by screws through the flange.

New Mold Controls Density

Breakage was encountered at the screw holes and more than 50 per cent of the production was scrapped because of this trouble. By specific gravity tests it was found that the density in the flange was lower than that of the body of the case. This indicated that an excessive amount of material was escaping at the last moment of closing of the mold. This lack of density resulted in such low strength in the flange that it was easily broken when pressure was applied on the assembly screws. The trouble was entirely eliminated by providing a new mold of the semipositive type. This type of design prevented the excessive escapement of material at the final squeeze of the mold and the density in the flange was held to the maximum. Lack of density or relatively high porosity would also affect the water absorption adversely. This in turn, would have a

Fig. 6 — Flock-filled phenolic was chosen for automobile aerial socket for shock resistance. Molding limitations for this material were overlooked, however, as also was fatigue from flexural strains in service

bad effect on the electrical properties.

Another instance involves a plastic part consisting of a collar or ring approximately 6 inches inside diameter by 2 inches deep with an average section thickness of ⅛-inch. When the molded parts were first produced they failed on the drop hammer test. A mold of the semipositive type was built and no further difficulty in meeting the drop hammer test was experienced. Since impact strength is involved in this case and since there is a wide range of phenolic impact materials available, the question might be raised as to the wisdom of spending money for a new mold rather than using a more shock-resistant material. The answer is that the extra strength materials are more expensive and this difference in material cost over one year's consumption would have been greater than the entire cost of the new mold.

Good Designs Give Satisfaction

A brief review of some excellent examples of good design which have insured satisfactory performance in service will serve to illustrate the possible advantages of molded plastic parts and their wide application to design.

The cellulose acetate steering wheel has been a tremendous contribution to safety in the modern automobile. Acetate is more flexible and tougher than the types of rubber compounds which previously were used for this purpose. The acetate rim is molded over a steel wire core which gives added strength. In collisions, the breakage of the older types of wheels from impact of the driver's body frequently caused serious damages to the driver. Under such impact the acetate wheel will bend rather than break.

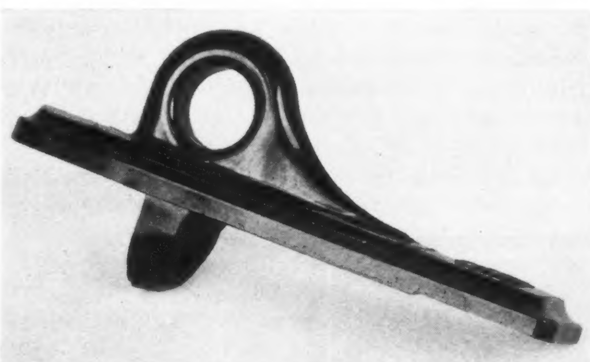


Fig. 7—Wood-flour-filled phenolic proved superior to fabric-filled materials because better strengths could be developed by controlling density and exactly filling complicated mold

Distributor caps are one of the first commercial applications of phenolic plastics. Designs have been modified to meet changing performance requirements of gasoline motors but phenolic materials are still being used successfully. These caps must be able to stand high dielectric fatigue under all weather conditions—cold, hot, wet and dry. Metal inserts embedded into the plastic must be accurately retained in position to insure satisfactory motor performance.

One of the most dramatic applications of thermoplastic methyl methacrylate is for the nose sections and "blisters" on bombers. The material is crystal clear and can be shaped to the streamline designs required. It has sufficient strength to support machine gun mounts and to withstand the pressures of power dives. It is considerably less fragile than glass and weighs less than half as much.

Incidentally this same material has been proposed for automobile windows and windshields to replace glass. The advantages over glass would be toughness, lighter weight and greater ease of forming into curved shapes, the latter satisfying the stylists' desire for greater streamlining. Although this plastic application appears to give very satisfactory performance on aircraft, its use on automobiles is questionable because of the abrasion problem from road mud. Methyl methacrylate, like all the other thermoplastic materials, is low in scratch hardness and would soon develop objectionable scratches from the frequent washing.

Utilizes Combinations of Molded Parts

Phenolic materials have been used for many years for spinning pots in the conventional process of making rayon. Aluminum was used originally, but a phenolic plastic was adopted about fifteen years ago. More recently the newer method of "spool spinning" rayon has required a wider use of plastics in the twisting bobbins and spinning reels. The latter are approximately twenty inches in diameter and built-up of several molded parts. When this new method of processing rayon was developed, a number of materials having the required resistance to chemical action were tried. Some could not be shaped satisfactorily and the weight of others was objectionable. An acid-resisting phenolic was selected because it had all of the desired properties.

There are of course thousands of other soundly engineered applications of plastics that are giving satisfactory performance in service. Each of these applications had its own peculiar design problems and there were good reasons for choosing the plastic in each case.

It is not possible within the limits of this paper to provide sufficient detailed information for the designing of a specific molded part. Rather, it has been the intention to bring out the importance of possessing considerable knowledge of the materials and their fabrication. The designer should be well acquainted with all of the materials before attacking a specific design problem and he should consult the detailed specifications of the various materials of the type or class selected just as he would if he were using metal alloys.

Designing Hydraulic Ser

By Christian E. Grosser

ONE of the most significant advances toward the objective of replacing human labor by mechanical or electrical power is the development of the devices known as servo-mechanisms. The British call them "slave-mechanisms," a term somewhat more suggestive of their function. This function, in a broad sense, is to supply power in proper form and accurately measured in quantity to carry out an assigned task with a minimum of direction or with no direction at all.

Unfortunately, knowledge of the operation and potentialities of servos is not as yet widespread. As a matter of fact the store of knowledge about their characteristics is still limited. Fundamental principles of behavior, however, have been well established and considerable experience has been acquired with a number of types of servo-mechanisms.

As a preliminary to the following discussion of hydraulic servo types in particular, it may be well to outline briefly the characteristics of servo in general and to refer to a number of applications so that the machine designer may have an index to their value as a tool.

Technically, the servo-mechanism has been defined as a power amplifying device in which the output driving element (called a servo-motor) is actuated by the difference (error) between the input to, and the output from, the servomechanism. The input referred to is the signal (or "command") imposed on the system at low power level, and the output is the response at the work-producing or driving element delivered at a magnified power level. That portion of the servo-mechanism which receives the input signal, compares it with the proper output quantity, and actuates the servo-motor so as to correct the error between the two, is called the "servo-controller." All servo-mechanisms, or as they are sometimes termed, "closed-cycle systems," may be described in these terms. However, some closed-cycle systems may be broken down into a number of servo-mechanisms, the output of one being applied to the input of another. *Fig.* 1 illustrates in diagrammatic form a general servo-mechanism.

Selection of the kind of component quantity of the output power is under no restriction provided a servo-controller can be found that will properly actuate the servo-motor which produces the component of power in question. Nor need the kind of signal input be restricted; it need not even be of the same kind as the output provided that the

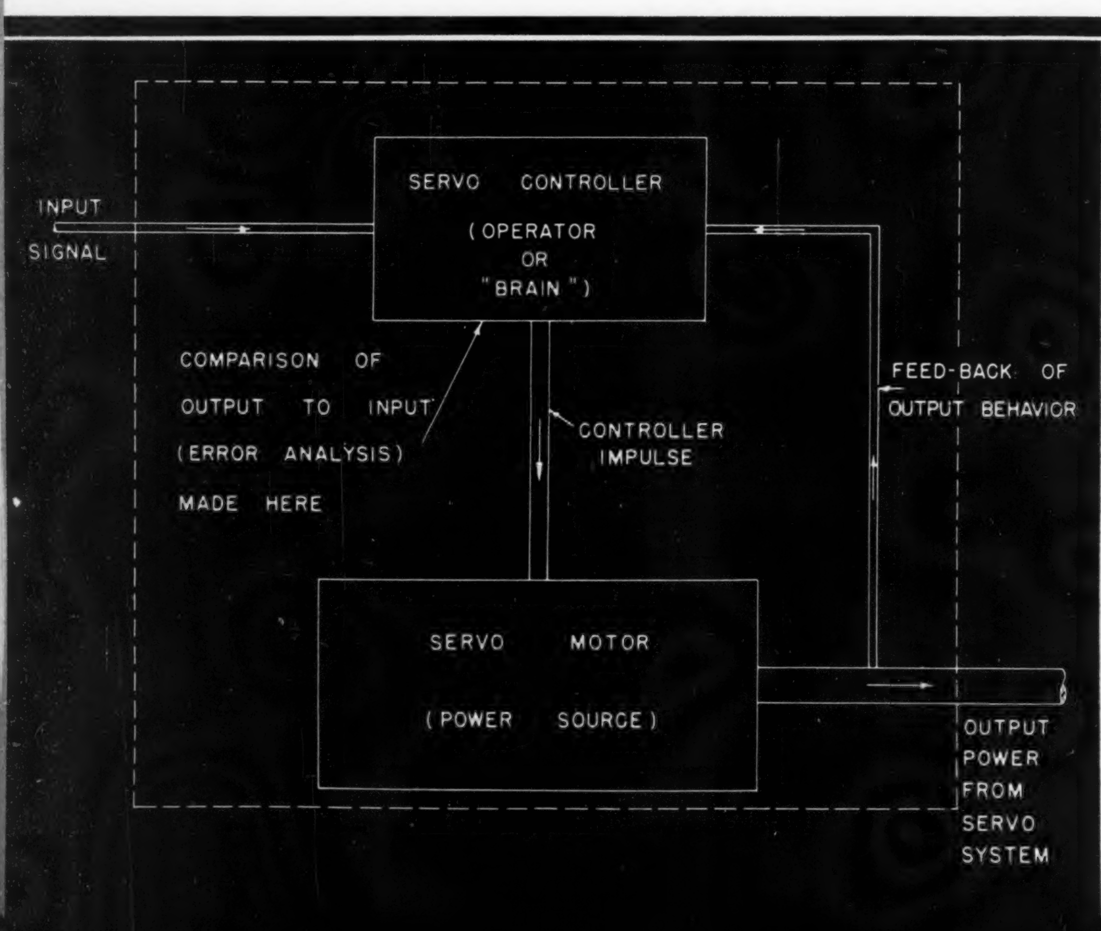


Fig. 1—Schematic diagram of a general hydraulic servo-circuit indicates how the driving element is actuated by the difference between the input and output of the servo-mechanism

...ircuits

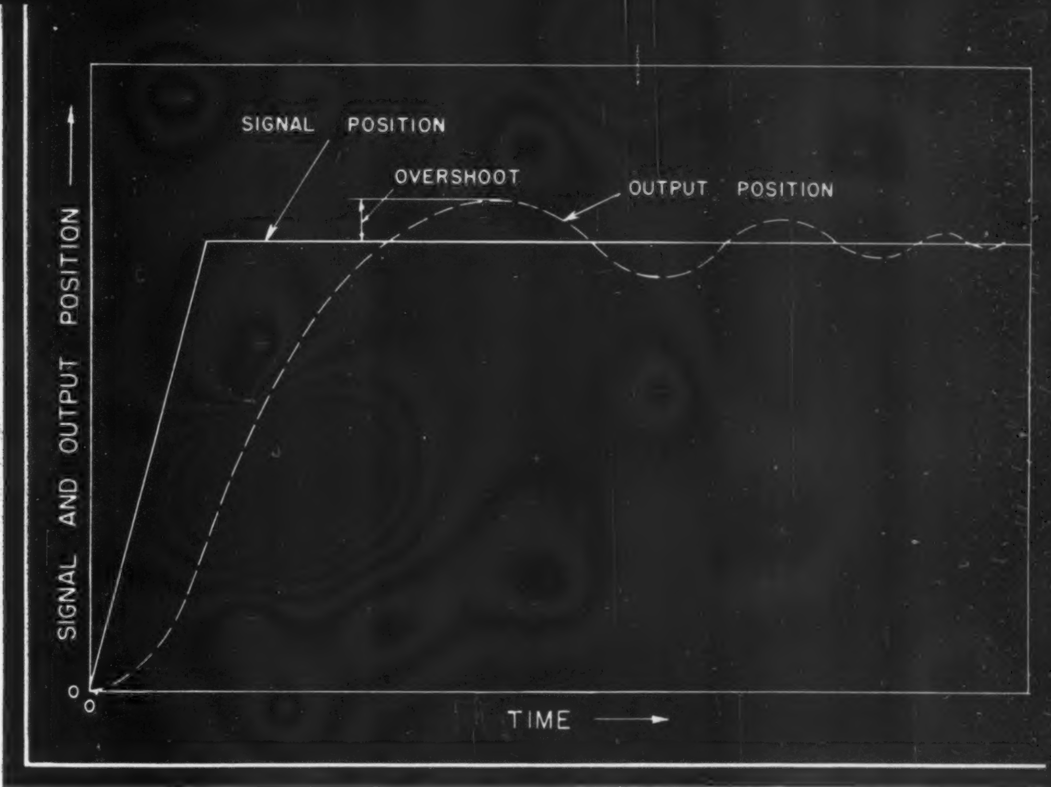


Fig. 2—Servo-circuits are prone to two types of error, lag and oscillation. Both may be minimized by various correction mechanisms

servo-controller can be made to recognize and maintain the desired relationship between it and the output quantity. For example, consider the servomechanism, or servo, used in steering airplanes. The pilot turns a dial as the signal to the steering servo to head the ship in a certain direction as indicated by the position of the dial shaft. The servo (a compass repeater) compares the dial setting with the existing direction of flight. If there is a difference it actuates the rudder to turn the ship toward the correct heading and continues to operate the rudder as long as error exists from the prescribed course.

Other applications of servos include: Compass repeater steering of ships, governing of turbine and engine speeds, ship stabilizers, thermostatic controls for furnaces or heating systems, pressure regulation in pumping systems, torque or power regulation in reeling operations, voltage or current regulation of generators. All these are characterized by the specification that a quantity of power component be maintained constant. This means that the input signal is set to a constant value, i.e., does not vary with time. Servos employed for this particular function are called "regulators."

From Instruments to Massive Machines

Of the more general applications of servomechanisms in which varying signals are applied, there are: Electrical or mechanical torque amplifiers in scientific instruments in which driving elements are required to follow the direction of measuring instruments with feeble power; hydraulic servo operation of massive machine tool tables or power presses in response to the motion of a small lever or dial; automatic profile following in milling, turning, and flame cutting.

In most of the applications mentioned in the foregoing the use, in the servo-mechanisms, of hydraulic components, such as servo-motors or controllers is possible and, in many instances, advantageous. The ideal servo would, of course, be one that would always maintain exact correspondence between the input signal and the output. This can hardly ever be achieved and so the servo designers' problem is to reduce the "error" to a practical minimum. Servo errors are of two kinds, "lag" and "oscillation." They are illustrated in *Fig. 2*.

Compromise between Lag and Overshoot

Lag may be reduced by increasing the speed with which the servo-controller actuates the servomotor. This degree of output reaction to error is called response ratio. As lag is decreased with a greater response ratio the servo has a tendency to "overshoot," or carry over beyond the condition of zero error on account of the inertia of load and servo-motor elements. The "overshoot" then constitutes a new error which the controller detects. It again causes the servo-motor to re-correct its output toward the condition prescribed by the input, which in turn may again cause an overshoot in the opposite direction. This process is oscillation. It may be reduced or eliminated by "damping," i.e., by providing means to absorb the energy of the vibration, and thereby be caused to die out quickly and completely, or to be limited in amplitude to an acceptably low value. A condition to be particularly guarded against in servo design is the introduction of influences which are called "negative damping." These increase the energy of oscillation instead of dissipating it. This condition, "instability," is characterized by the vibration of the output at an amplitude of error that increases with time until failure occurs in mechanism.

Various other principles have been discovered by which servo errors may be minimized, notably "error derivative correction" and "error integral

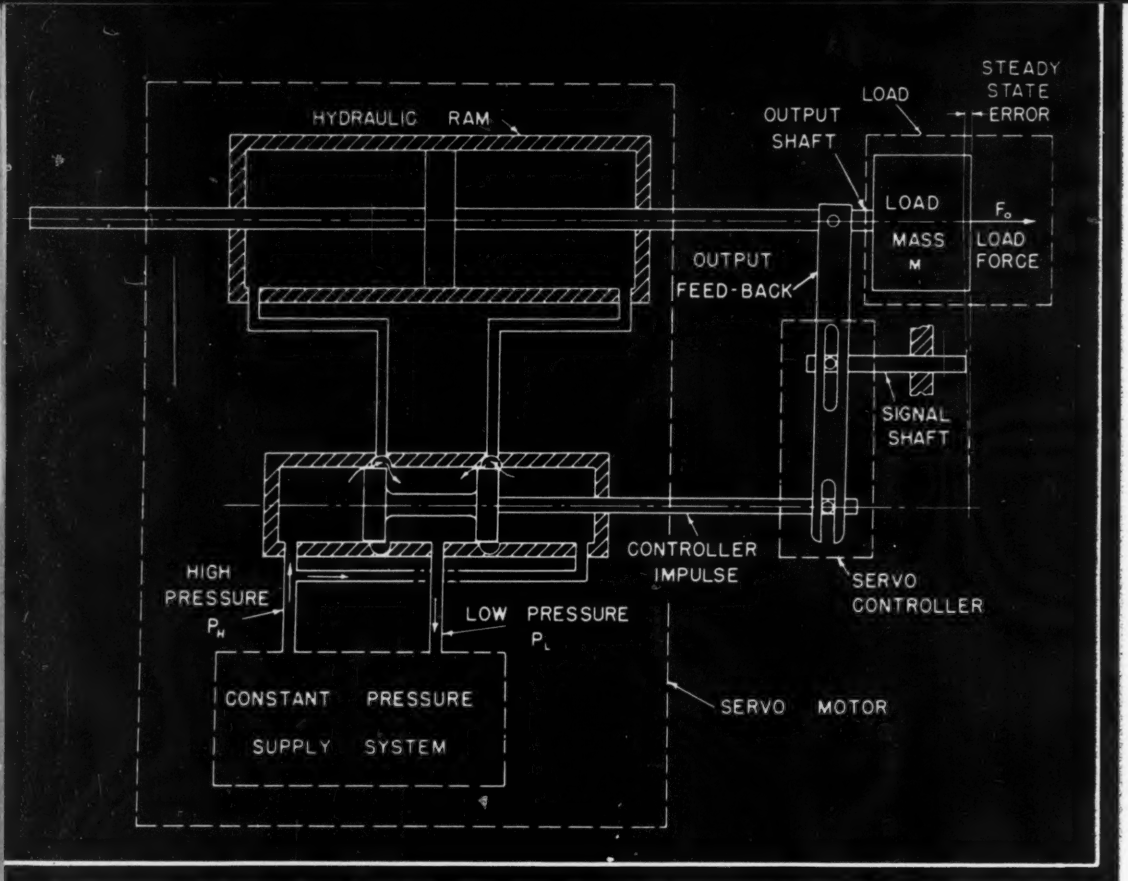


Fig. 3—Compensation for stationary or steady-state error requires a slight displacement of four-way valve

correction." Such corrections are introduced by certain kinds of servo controllers in addition to the error correction itself. A detailed discussion of error causes and their correction in servo systems in general is not within the scope of this article. For such information the reader is referred to more comprehensive treatments listed under the appended references.

How To Vary Output Speed

If a servo is to be used to supply some form of mechanical power such as the motion of a machine element to follow a varying signal output, an essential characteristic of the servo-motor is a variable speed output. Both electrical and hydraulic power equipment fit into this requirement nicely since their speeds may be uniformly varied with ease. In the case of electrical motors used as servo-motors, the servo-controller can usually operate only on the potential applied to the motor. Since this is not the only important factor affecting speed, the control of speed is indirect. In the hydraulic transmission, on the other hand, the flow of the liquid to a fluid motor may be made the principal determining factor of speed, and may be regulated directly by a controller mechanism; as, for instance, by varying the stroke of a variable displacement pump or through the use of a metering valve in the circuit. As an alternative, the pressure of the hydraulic circuit may be caused to vary by a servo-controller through throttling valves.

Hydraulic servo-motors have an inherent advantage in servo-operation because of their relatively low inertia, low weight, and low space requirements for large power transmitting capacity. On account of low output inertia a high degree of accuracy may be obtained with relatively simple controllers. A hydraulic servo may also be damped effectively and easily.

As an example of the kind of analysis which may be made of hydraulic servo-mechanisms to obtain an index of their performance and to judge the most effective proportions of a design to perform specific functions, a simple and rather commonly used type of translatory servo will be considered. *Fig.* 3 shows the circuit arrangement. The servo-motor is composed of a constant-pressure hydraulic generator connected through a four-way throttling valve to a hydraulic ram. (In the diagram the hydraulic cylinder is shown as being analogous to any form of hydraulic motor.) The servo-controller is a mechanical linkage which actuates the four-way valve in proportion to the error between the position of the output shaft and the input, or signal shaft. Any change in position or speed of the input signal shaft will be followed by an approximately proportional motion of the output shaft. *Fig.* 3 shows the system in equilibrium against a load force F_0. A small "stationary error" exists because of the presence of this applied force, necessitating a slight displacement of the four-way valve in order that a higher pressure may apply to the right side of the ram piston than that applied to the left side.

Lag or Overshoot Determined

In *Fig.* 4 the system is shown immediately after the signal shaft has been displaced to the left. It will be noted that the controller linkage has now opened the four-way valve by a large amount, admitting high pressure to the right side of the ram

piston and low pressure to the left. Thus motion of the ram is beginning in the direction to reduce the error between output and input positions.

Prescribing now that the input shaft be held stationary, it is desirable to determine how quickly the output arrives at its newly prescribed position, and whether it will overshoot and oscillate after arrival. If it be assumed that there is negligible leakage past the ram piston and across the lands of the four-way valve we may write the equation that the sum of the pressure drops along the oil circuit starting from and returning to, the pressure generator or pump is equal to the constant pressure differential maintained across the terminals of the pump.

$$P_h - P_1 = \frac{M}{a_m}\frac{d(v_m)}{dt} + \frac{F_o}{a_m} + \frac{\rho v_1^2}{2} f \frac{L}{d} +$$

$$\left.\begin{matrix}\text{Pressure}\\\text{maintained}\\\text{by pump}\end{matrix}\right\} = \left\{\begin{matrix}\text{Pressure}\\\text{difference}\\\text{in ram}\end{matrix}\right\} + \left\{\begin{matrix}\text{Straight}\\\text{pipe}\\\text{loss}\end{matrix}\right\} +$$

$$\frac{\rho}{2} v_1^2 n K + \frac{\rho v_1^2}{C_d^2}\left(\frac{a_1^2}{a_0^2} - 1\right) \quad \text{(1)}$$

$$\left\{\begin{matrix}\text{Entrance loss at}\\ n \text{ enlargements}\\\text{or contractions}\\\text{in pipe circuit}\end{matrix}\right\} + \left\{\begin{matrix}\text{Pressure drop}\\\text{through the two}\\\text{valve orifices}\end{matrix}\right\}$$

where,

- M = total effective mass of load, servomotor, and accelerated liquid
- v_m = velocity of output shaft
- a_m = area of ram piston
- F_o = load force in direction shown
- ρ = mass density of the liquid
- v_1 = average velocity of liquid in connecting lines
- f = pipe friction factor, the usual function of Reynolds' number = .02, (approx.)
- L = total length of connecting pipes and passages
- d = connecting pipe or passage diameter
- n = number of entrance and exit losses
- K = average loss factor for enlargements and contractions in circuit = .75, (approx).
- C_d = coefficient of discharge at annular valve orifice .6, (approx.)
- a_1 = connecting pipe or passage area
- a_0 = valve opening orifice area

Noting that if the assumption regarding leakage holds,

$$v_1 a_1 = v_m a_m \quad \text{(2)}$$

and substituting for v_1 in Equation 1,

$$P_h - P_1 = \frac{M}{a_m}\frac{d(v_m)}{dt} + \frac{F_o}{a_m} + \frac{\rho}{2} v_m^2 \frac{a_m^2}{a_1^2}\left[f\frac{L}{d} + nK + \frac{2}{C_d^2}\left(\frac{a_1^2}{a_0^2} - 1\right)\right] \quad \text{(3)}$$

Transposing, this becomes

$$\frac{d(v_m)}{\left(P_h - P_1 - \frac{F_o}{a_m}\right) - \frac{\rho}{2}\frac{a_m^2}{a_1^2}\left[f\frac{L}{d} + nK + \frac{2}{C_d^2}\left(\frac{a_1^2}{a_0^2} - 1\right)\right] v_m^2} = \frac{a_m}{M} dt \quad \text{(4)}$$

This cannot be integrated as it stands since a_0 is a function of v_m by the movement of the valve as the output shaft moves. However, if a_0 is assumed to be constant over a short period of time Δt, the integration may be carried out with reasonable ac-

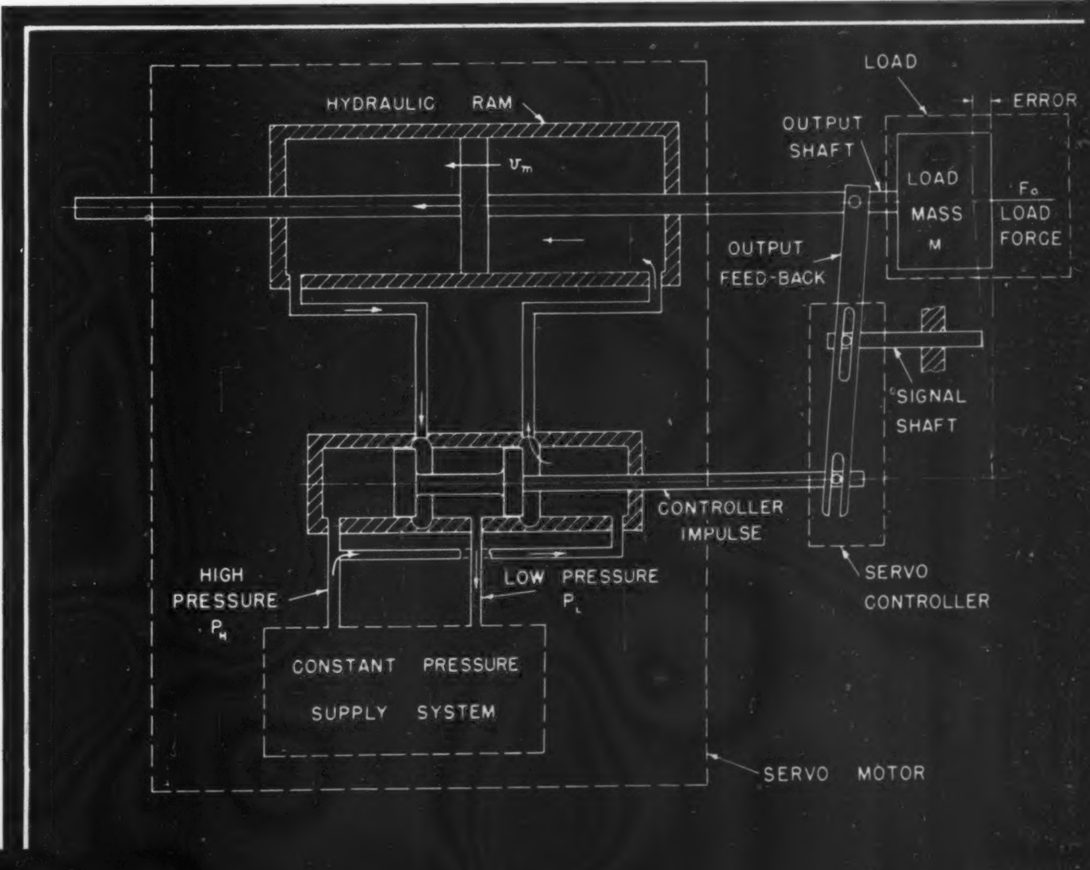


Fig. 4—This circuit is the same as that shown in Fig. 3 but with the four-way valve opened in response to a displacement of the signal shaft

curacy over this short time interval (a small fraction of a second).

Equation 4 is then of the form

$$\frac{d(v_m)}{A - Cv_m^2} = \frac{a_m}{M}\,dt$$

the solution of which is

$$\left[\frac{1}{2\sqrt{AC}} \log \frac{Cv_m - \sqrt{AC}}{Cv_m + \sqrt{AC}}\right]_{v_{mi}}^{v_{m(i+1)}}$$

$$= \frac{a_m}{M}\,(t_{(i+1)} - t_i) = \frac{a_m}{M}\,\Delta t \quad \ldots\ldots\ldots (5)$$

where A and C are the constants of the denominator of the left side of Equation 4, i represents the beginning of one time interval and $(i + 1)$ the beginning of the next time interval.

Now from Equation 5, $v_{m(i+1)}$ at the end of any given time interval, i.e., at $t_{(i+1)}$ may be determined in terms of the velocity v_{mi} established at the end of the previous time interval, i.e., at t_i. The assumed constant value of a_0 introduced during each evaluation of Equation 5 may be estimated from the position of the output shaft at the end of the previous time interval and from the corresponding position of the four-way valve. The successive positions of the output shaft may be calculated by multiplying Δt by the average velocity over the interval and may then be plotted as in *Fig.* 5. From this plot may be obtained the time taken by the output shaft to reach the position prescribed by the signal and the initial velocity of overshoot.

The four-way valve at this point is now on dead center and the assumption regarding absence of appreciable leakage past the valve lands, no longer holds since the length of the seal on the lands in a valve as shown is now zero. New equations describing the dynamics of the system during and beyond the point at which the valve crosses dead center might now be written. However, they are complicated and will not be attempted here.

Larger Cylinder Reduces Overshoot

Furthermore, it is usually sufficient to carry the calculation this far in a design, and to judge from the time taken to make the correction and from the initial velocity of overshoot, whether the servo selected is adequate. If the initial velocity of overshoot is high it is best, in order to avoid extended oscillation, to select a somewhat larger servo-cylinder, or to reduce the pressure supplied by the generator. The latter procedure will, of course, increase the time necessary to make correction, and increase the stationary error under the imposed force F_0. In no case can the supply pressure differential $P_h - P_1$ be carried down to the value F_0/a_m since then, obviously, no allowance is made for pressure drops in the piping or for acceleration of the masses involved and the servo will not respond. A satisfactory value for the pressure differential can usually be found in the neighborhood of $2F_0/a_m$, although this should be checked in each case by applying Equation 5.

It is important to point out that this type of

(*Concluded on Page* 128)

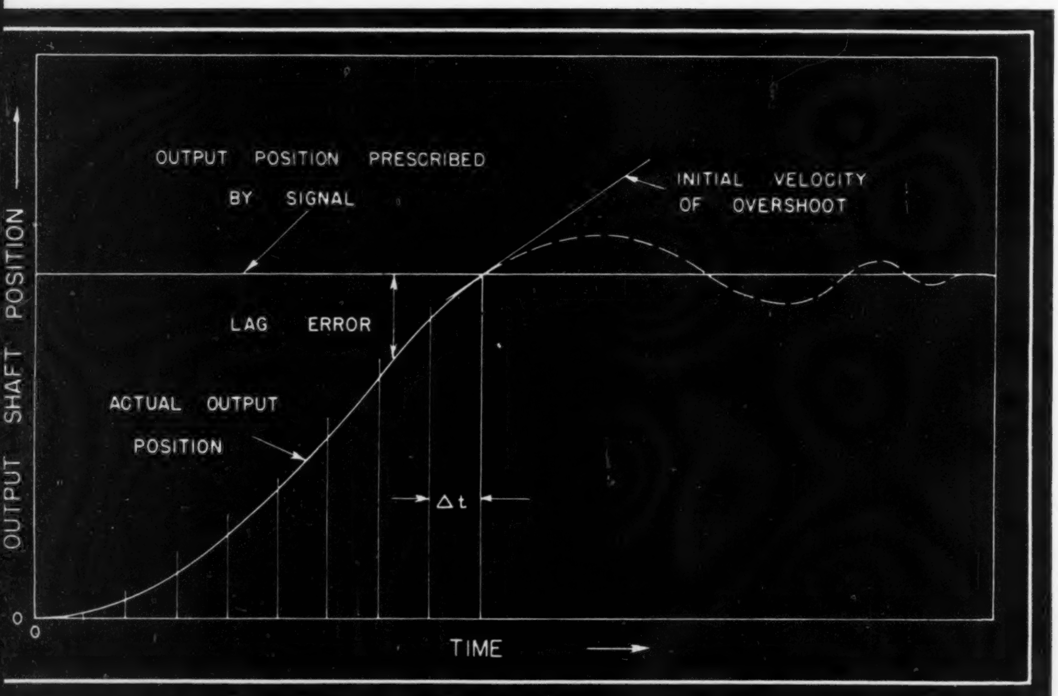
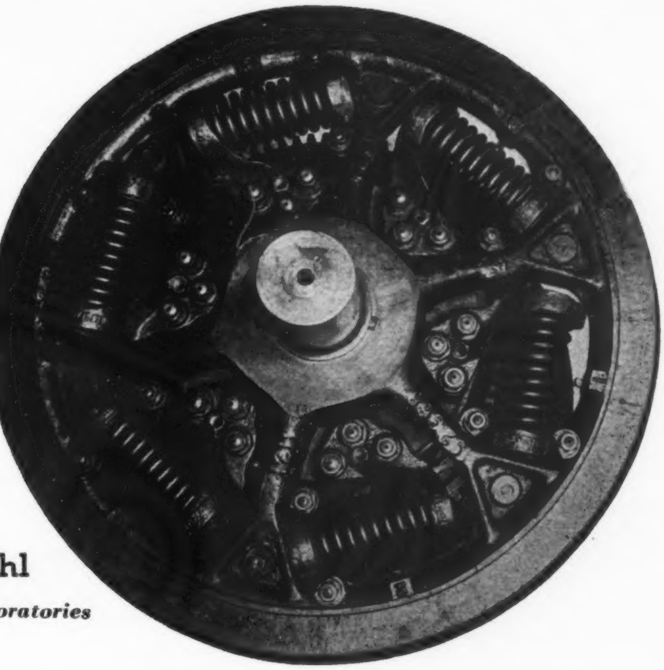


Fig. 5—Graph permits the determination of the time lag for positioning the output shaft and also the initial velocity of overshoot



By A. M. Wahl

Westinghouse Research Laboratories

Fig. 1—Drive springs in torsional coupling absorb maximum shock in limited space available

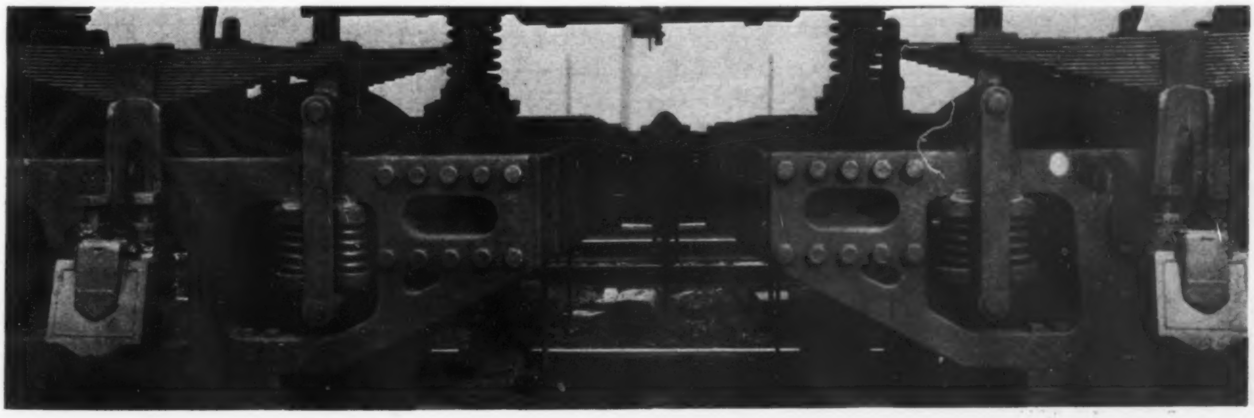
Combining Maximum Spring Deflection with Minimum Space

DESIGNING a helical spring to obtain maximum deflection at a given load and stress with the further requirement that the spring must fit into a given space is a problem which frequently arises. This is equivalent to the problem of designing springs to store a maximum amount of potential energy per unit volume of space occupied. Typical applications where space limitations are controlling factors are shown in *Figs*. 1 and 2.

A logical approach to the determination of the optimum spring index for this condition is to assume that the spring is a compression spring and that the coils just touch when the stress reaches its maximum permissible value. The amount of energy stored is then calculated for various spring indexes. It will be found that in general, for a given volume of space occupied by the spring, the amount of energy which can be stored at a given peak stress will be a maximum at a definite spring index.

This optimum value of the index will be different, depending on whether the spring is subject to a static or to a variable load. The method of approach

Fig. 2 — Equalization anchor springs for electric locomotive are subject to space limitations



is somewhat similar to that used by J. Jennings[1]. However, the method used in this article differs from that used by Jennings in that a distinction is made between static and variable loading; in addition the usual helical spring deflection formula rather than the Wood formula is used since tests show the former to be accurate[2].

The potential energy U stored in a helical compression spring with load P and deflection δ is

$$U = \frac{1}{2} P \delta \qquad (1)$$

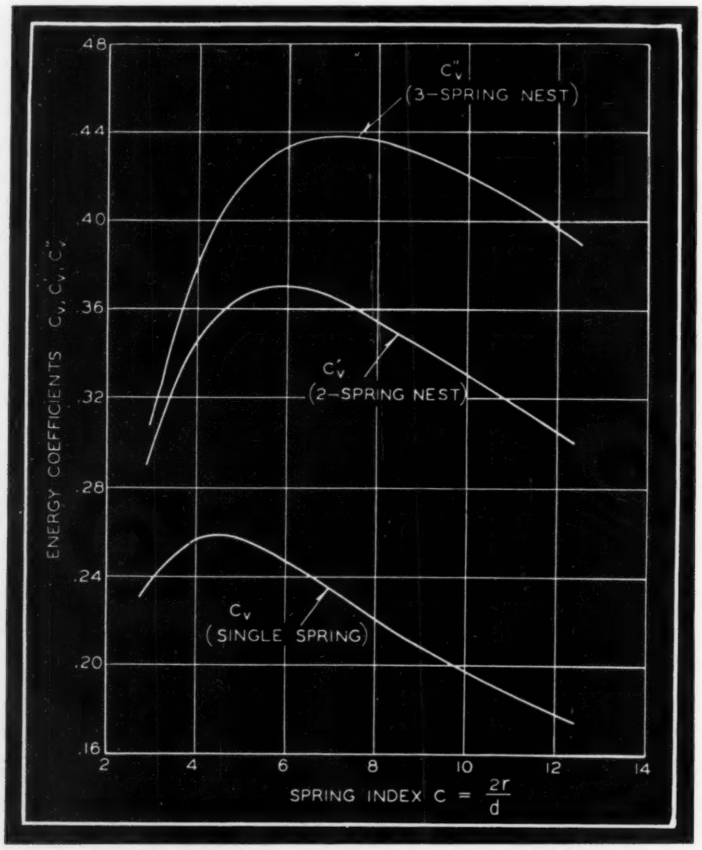


Fig. 3—Energy coefficients for variable loads

The deflection δ of the spring (from the usual deflection formula) is

$$\delta = \frac{64 P r^3 n}{G d^4} \qquad (2)$$

where r = mean coil radius, n = number of active coils, G = shear modulus of elasticity, and d = wire diameter.

Substituting Equation 2 in Equation 1, the energy stored becomes

$$U = \frac{32 P^2 r^3 n}{G d^4} \qquad (3)$$

[1] *Engineering*, page 134, Aug. 15, 1941.
[2] *"Factors Affecting Accurate Spring Selection"*, MACHINE DESIGN, Aug., 1939.
[3] *"General Considerations in Designing Mechanical Springs"*, MACHINE DESIGN, Feb., 1938.
[4] A slight error is made by neglecting the helix angle, but this is of no importance in the present instance.

If the load acting on the spring is variable (for instance, that on an automotive valve spring), the peak torsional stress in the spring, as a first approximation, may be used as measure of the load-carrying ability. This stress is

$$S = \frac{16 PrK}{\pi d^3} \qquad (4)$$

where the factor K takes into account effects of curvature and direct shear. This factor is

$$K = \frac{4c-1}{4c-4} + \frac{.615}{c}$$

where c = spring index[3]. Solving Equation 4 for P and substituting in Equation 3,

$$U = \frac{\pi^2 r n d^2 S^2}{8 G K^2} \qquad (5)$$

This energy must be stored in a volume of space bounded by the outside diameter of the spring and its ends. The criterion for space required will therefore be taken as the volume of a cylinder with a diameter equal to the outside coil diameter, $2r + d$, and a length equal to the active length, nd, of the spring when the latter is fully compressed.[4] This volume is

$$V = nd \frac{\pi}{4} (2r+d)^2 = \frac{nd^3 \pi}{4} (c+1)^2 \qquad (6)$$

which may be written as

$$nd^3 = \frac{4V}{\pi (c+1)^2} \qquad (7)$$

Substituting Equation 7 in 5, we obtain:

$$U = C_V \frac{S^2 V}{4G} \qquad (8)$$

where C_V is a constant depending on the spring index c. Thus

$$C_V = \frac{K^2}{\pi c} \frac{1}{(c+1)^2} \qquad (9)$$

This equation shows that for a given volume of space occupied and a given stress, the energy stored depends only on the energy coefficient C_V which in turn depends only on the spring index. Values of C_V are plotted against spring index c in the lower curve of *Fig.* 3. This curve shows that for *variable* loads and for a single spring the maximum energy is stored in a given space and at a given peak stress if the spring index is between 4 and 5.

Where the load is *static* or repeated only a few times during the service life of the spring, the indications are that curvature effects (but not those due to direct shear) may be neglected in calculat-

ing the stress[5]. In this case the stress is given by:

$$S_S = \frac{16\,Pr}{\pi d^3} K_S \qquad (10)$$

where

$$K_S = 1 + \frac{.5}{c}$$

The factor K_S takes into account the stress augmented by direct shear. Using this equation instead of Equation 4 and proceeding in a way similar to before, the energy stored becomes

$$U = C_S \frac{S_S^2 V}{4\,G} \qquad (11)$$

where

$$C_S = \frac{\pi c}{K_S^2} \frac{1}{(c+1)^2} \qquad (12)$$

Plotting C_S as a function of the spring index c, the lower curve of *Fig.* 4 is obtained. This curve indicates a maximum value of C_S at the smallest practical spring index, about 3. This means that where static loads are involved, maximum energy storage using only one spring will be obtained in a given space by using the smallest practical value of the index.

Spring Nest Increases Loading

The energy stored within a given space may be increased by using a spring nest of say two or three springs telescoped one inside the other. For maximum energy storage, the solid lengths of the springs comprising the nest should be the same. Assuming a nest composed of two springs, this means that

$$n_1 d_1 = n_2 d_2 \qquad (13)$$

In this equation and those following, the subscripts 1 and 2 refer to the outer and inner springs of the nest respectively. In addition, it will be assumed that the total deflection of each spring at any given load is the same. This yields the following condition

$$\delta_1 = \delta_2 \qquad (14)$$

In terms of stress for variable loading, the deflection, using Equations 2 and 4 is

$$\delta_1 = \frac{4\pi r_1^2 S_1 n_1}{G d_1 K_1} \quad ; \quad \delta_2 = \frac{4\pi r_2^2 S_2 n_2}{G d_2 K_2} \qquad (15)$$

In terms of the spring indexes c_1 and c_2 these may be written as

$$\delta_1 = \frac{\pi c_1^2 n_1 d_1 S_1}{G K_1} \quad ; \quad \delta_2 = \frac{\pi c_2^2 n_2 d_2 S_2}{G K_2} \qquad (16)$$

Assuming the same maximum stress in each spring, i.e., $S_1 = S_2$, then since $n_1 d_1 = n_2 d_2$ from Equation 13 the spring indexes c_1 and c_2 (and hence also the curvature correction factors K_1 and K_2) should be the same if $\delta_1 = \delta_2$. When the spring indexes are the same, the energy coefficients C_V (which depend only on the indexes) will be the same for both springs. Using Equation 8 this means that the total energy stored will be given by

$$U = C_V \frac{V_1 S_1^2}{4\,G} + C_V \frac{V_2 S_2^2}{4\,G} \qquad (17)$$

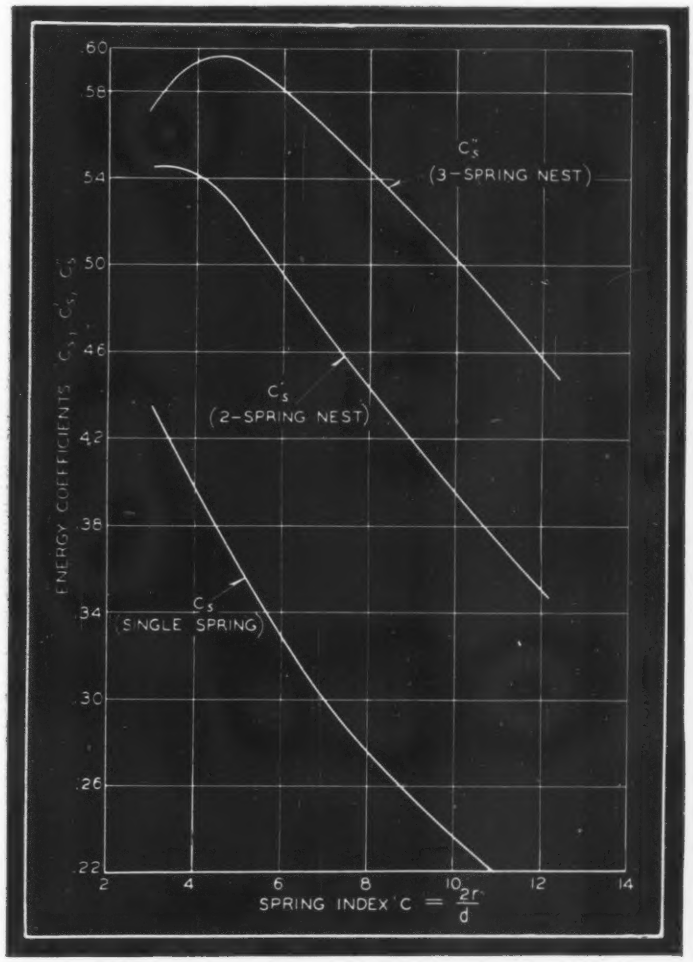


Fig. 4—Energy coefficients for static loads

where V_1 and V_2 are the volumes enclosed by the outer and inner springs, respectively, when compressed solid. Since $S_1 = S_2 = S$, this equation may be written

$$U = C_V \frac{V_1 S^2}{4\,G}\left(1 + \frac{V_2}{V_1}\right) \qquad (18)$$

If it is now assumed that the outer diameter of

(*Continued on page* 102)

[5] "*Calculating Springs for Static Loading*", MACHINE DESIGN, June, 1941 contains a further discussion of this point.

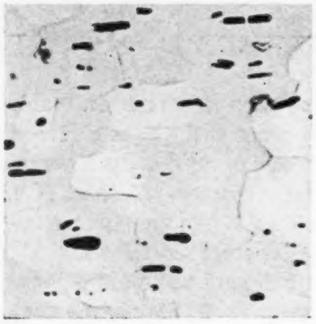


Fig 1 — Microstructure of 2S-O aluminum sheet

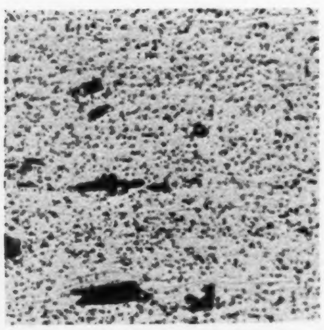


Fig. 2 — Microstructure of 3S-O aluminum sheet

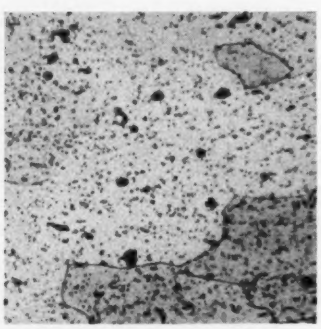
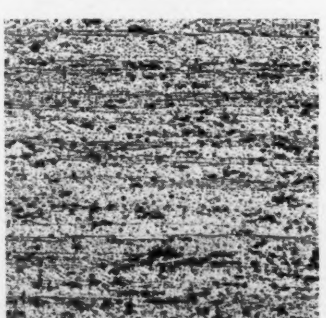


Fig. 3 — Microstructure of 52S-O aluminum sheet

Fig. 4 — Microstructure of 24S-O aluminum sheet



How Metallurgy A

By Kenneth D. Moslander

INTEREST of the design engineer in the metallurgical aspects of deep drawing and pressing is confined almost exclusively to results. Foremost among these "results" are cost, quality and efficiency of the finished part. These he obtains by specifications directed to the stampings manufacturer, such specifications including: Kind of metal, usually dictated by the service for which the part is intended; surface finish, controlled by considerations of appearance or the ability to provide a satisfactory base for platings, resins, lacquers, etc.; physical properties requisite in the application of the part.

Aside from these and possibly a few other related "result" considerations, what interest does the design engineer have in the metallurgy involved? To what extent does grain size and directionality, plastic range, chemical composition, heat treatment, cold working, etc., constitute a problem for him? An attempt to indicate the answers to these questions will be made, though it would be presumptuous to prescribe definite and arbitrary laws and formulas on so controversial and even contradictory a subject. However, those metallurgical factors which affect the finished part and which the designer should consider in drawing his specifications will be discussed.

As concisely stated by J. D. Jevons[1]: ". . . in view of the lack of knowledge concerning many of the properties which at the present time are considered to be of primary importance, and of the inability of many commonly applied tests to reveal the complete suitability or otherwise of sheet tested, adequate specification of deep-drawing quality metal sheet is a matter of great difficulty." Further, "the fact that a satisfactory specification of accepted type cannot at the present time be drawn up seems to occasion chagrin . . . . "

Kind of Metal Dictated by Use

In the light of the foregoing it is not easy to go further. But, even if a "satisfactory specification" cannot be written at once on the finished drawing, an understanding of the fundamental metallurgy will permit of a far more intelligent first approximation.

Whether a part is to be made of steel, brass, aluminum, nickel or other material, is a matter of the individual requirement of the application and can be passed over lightly at this time. Such a decision may be based on resistance to corrosion, strength, rigidity, weight, service, etc., and is usually made before the part is finally designed and without regard to the press-working difficulties which may be introduced.

Generally, a pure metal lends itself more readily to being drawn and formed than a metal with uncombined alloying constituents. In steel, the presence and distribution of carbides exercise a profound influence on deep-drawing qualities. Hence, low carbon, low alloy (killed or rimming) steel is used practically exclusively in drawn parts. (Recently, however, stainless steel is being used increasingly.) Similarly, alpha brasses containing one-third or less zinc are best adapted to the cold-working processes.

Minor constituents such as phosphorous in steel or iron in brass which

[1] *The Metallurgy of Deep Drawing and Pressing*, Page 539, 1940, John Wiley & Sons Inc.

ects Deep-Drawing

may have a pronounced effect on drawing properties are best determined and controlled by the materials supplier. For this reason it is essential that the supplier be provided with, in addition to a blueprint of the part, a statement as to its use and subsequent treatment, i.e., heat treatment, plating, enameling, etc.

Surface finish, particularly in the case of steel, is vital. "Mirror finish" cold-rolled sheet may have definite advantage from the appearance standpoint but provides working difficulties because of its inability to be "wet" by die lubricants. On the other hand, oxide scale on hot finished sheet constitutes a serious hazard for punches and dies in press forming. The answer in either case is the specification of a final pickling process which, incidentally, also improves the "tooth" for varnishes, lacquers or enamels.

Equally or more important than grain size is the uniformity or regularity of the size of grains or crystals as indicated in the several illustrations which will be referred to in more detail later. Basically this is true because of the controlling effect of grain size on such drawing properties as ductility, tenacity, hardness, plastic range, etc. Generally an increase in grain size results in an increase in ductility and plastic range but a decrease in tenacity, tensile strength and hardness. Also, an inordinate increase in crystal size aggravates the phenomena of "orange peel" or pebble-grained surface in highly strained metal.

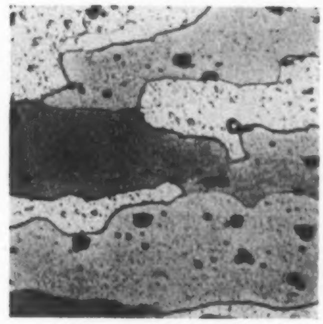


Fig. 5—Microstructure of 24S-T aluminum sheet. Hardening constituents are retained in solution or are of submicroscopic size

Fig. 6—Right—High-purity aluminum surface layers on 24S-T sheet show the diffusion zone resulting from molecular migration

Grain Size Compromise Necessary

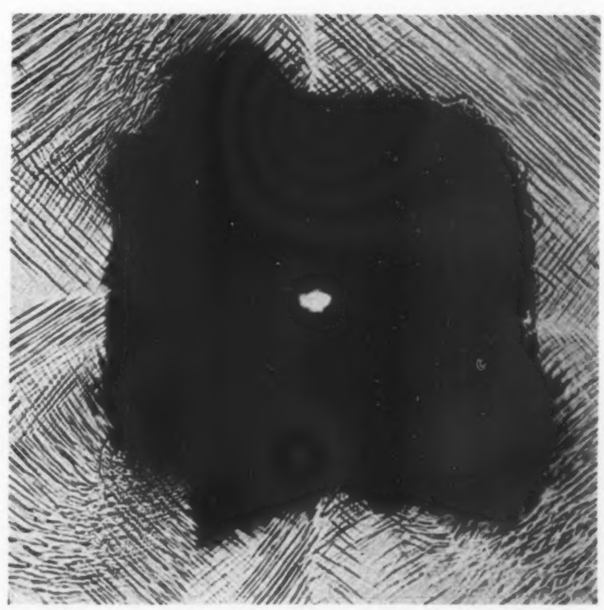
Metal brittleness, although resulting more directly from the relative orientation of the crystals, is far less likely to occur in metal of small grain size. Ridges or line depressions known as stretcher lines, the former resulting from compression and the latter from tension stresses in the metal, are a consequence of uneven flow of the metal in drawing. The smaller the grain size, the greater is the tendency toward the formation of such lines.

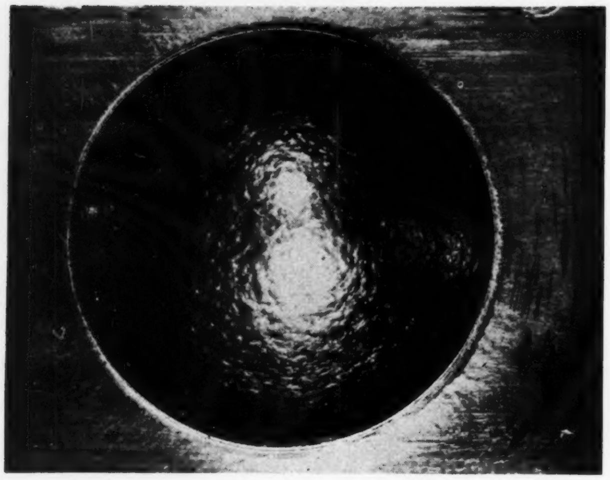
It therefore is evident that, in order to obtain the necessary ductility of metal and a satisfactory surface of the finished part, a compromise between maximum and minimum grain size must be resorted to. A practical range is between .035 and .050 millmeters. Grain size in the direction of, or even slightly smaller than, the lower limit is used in thin sheet.

Thus it will be seen why any extensive variation in crystal size is unsatisfactory for drawing purposes. If a section of material consists of grains considerably larger than in other sections, the larger grain section will be more ductile but less strong. Hence, when stressed, this section will neck down and, because it is unable to transmit sufficient stress to strain the smaller grain section, will fail.

Work-hardening is a property without which a

Fig. 7—Below—Characteristic slip lines in aluminum were formed by a rockwell impression on a polished metallographic surface





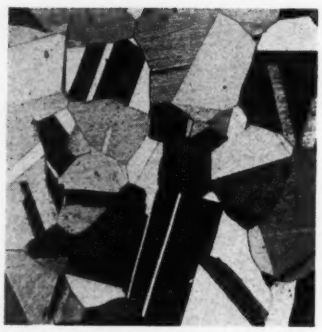


Fig. 8—Above—Excessively coarse-grained metal results in the typical orange peel surface

Fig. 9—Left—Characteristic appearance of the microstructure of deep drawing cartridge brass

metal could not be drawn. As a metal necks down in drawing, the stress at the elastic limit of the reduced portion must increase in order to transmit sufficient stress to strain the unreduced metal. If the stress does not increase sufficiently, necking down will be followed directly by failure.

In the previous article of this series (December, 1940) recourse was taken to the analogy of modelling clay to drawing operations. This material, of course, does not work harden. Therefore, while such an analogy is satisfactory for demonstrating tendencies toward wrinkling and thinning it breaks down from the metallurgical viewpoint. In this case, the stress at the "yield point" remains constant. Hence as the section reduces, the total load to produce deformation also reduces, with the result that all flow of material takes place in the initially most-strained portion, encouraging early failure.

Lead work-hardens but little and therefore does not draw well. Pure metals, generally, exhibit a similar tendency. High-purity aluminum, despite high ductility, does not work-harden adequately. This is the chief reason why the depth of draw in this material in one operation must usually be less than, for example, steel or brass.

On the other hand, excessive work-hardening is equally detrimental to drawing. Stainless steel, for example, which is lately being much used, work-hardens rapidly as evidenced by its high slope on a true stress-strain curve. This condition necessitates frequent annealing in deep drawing and, because of increased hardness and strength, subjects the dies to severe wear, further contributing to high cost.

Directionality of grain is an important specification and should be considered carefully by the designer in his layout. In the November article of this series the importance of directionality in bend-forming was pointed out. Metallurgically it is a consequence of an orientation of grains such that the crystal boundaries are more or less aligned. These boundaries constitute an incipient slip plane and result in the material being more ductile in one direction than the other. In drawing operations this is evidenced by a consistent direction of tearing when a piece is strained to failure, or by the formation of "ears" in drawn shells. The former illustrates proneness to failure due to low ductility; the latter an increase in cost due to trimming requirements as well as an increased likelihood of wrinkling if pad pressures and process are adjusted to draw the unsatisfactory metal.

The foregoing considerations are all, within prac-

TABLE I

Nominal Composition of Aluminum Alloys

Alloy	Alloying Elements, (per cent)*				
	Cu	Si	Mn	Mg	Cr
2S					
3S			1.2		
52S				2.5	.25
17S	4.0		0.5	0.5	
24S	4.5		0.6	1.5	
53S		.7		1.3	.25

*Aluminum and normal impurities constitute remainder.

tical limits, controllable. However, their control may and usually does result in a cost increment of the material. In specifying material for drawn parts it is false economy to use a cheap material even though its chemical composition may be identical with that of a somewhat more expensive one. Excessive amounts of scrap, wear on the tools, higher press power requirements, unsatisfactory surface finish are but a few of the elements which will erase rapidly any saving that might be hoped for by using cheaper metal.

Often too, a higher material cost may permit the elimination of one or more operations in the drawing of a part. If an interstage annealing can be avoided or if a drawing operation can be dispensed with, a cost increment in material is easily justified. This question should be discussed in detail with the stampings manufacturer or with the

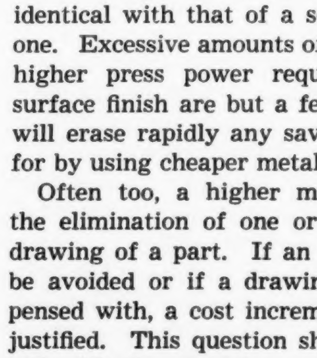


Fig. 10—This microstructure results from extremely high cold-working of the cartridge brass shown in Fig. 9

Fig. 11—Grain size of this magnitude results in the orange peel effect shown in Fig. 12

material supplier with whom he is in constant consultation. Their recommendations, based as they are on practical experience, are in the final analysis of far more value than any theoretical discourse on metallurgy could be.

The foregoing discussion is intended to provide a broad picture of the metallurgical aspects of the subject which are, in general, applicable to all materials which lend themselves to being drawn. It will be of further interest to discuss in some detail several specific materials widely used in the process in order to provide a more concrete basis for material specification by the engineer.

TABLE II

Capacity of Various Drawing Presses

Type of Press	Capacity of Press (tons)	Max. Dia of Shell (inches)	Max. Blank Thickness (inches)	Max. Depth of Draw (inches)
Double-Action Toggle	63	8⅞	.051	4¾
Double-Action Toggle	125	14	.102	8½
Double-Action Toggle	148	15½	.102	9½
Double-Action Toggle	282	23	.125	10
Double-Action Toggle	300	27	.125	10
Double-Action Toggle	360	25	.125	14½
Double-Action Toggle	400	30	.156	12
Double-Action Toggle	550	30	.187	17
Double-Action Toggle	500	37	.250	14
Hydraulic	1300	49	.3125	24

In addition to the quality of formability the selection of a material for a deep-drawn part depends upon a number of other factors. Despite the generally good corrosion resistance of aluminum, even in this case certain alloys are better suited to some applications than others. For example, exposure to marine atmosphere is more satisfactorily resisted by such alloys as 52-S or 53-S shown in TABLE I. In other cases, an optimum combination of chemical as well as physical properties is essential. In such applications the choice of a material with the proper temper must be considered. This can only be done by reference to the design and by estimating the amount of reduction taken in the total draw when no annealing operation is used, or after the last anneal when one or more anneals are used. A material is then selected whose temper is sufficiently lower than that desired in the final part so that the amount of cold working incurred in the production of the final part will result in the degree of temper desired.

Presses required for drawing aluminum are the same as those used for other metals. Their size and capacity depend upon the diameter of the blank, thickness of metal, shape of the drawn item, and temper of the material. Although the press used is not essentially a metallurgical problem it does have an important bearing on the cost of the part. For this reason TABLE II is included. Generally the cost per piece varies directly with the press capacity. TABLE II, therefore, should provide a criterion of the relative values of this element of the total part cost.

Aluminum alloys generally used for the production of drawn shapes are 2-S, 17-S, 24-S, 52-S, and 53-S, the physical properties of which are given in TABLE III. The 2-S, 3-S and 52-S alloys are generally referred to as "common alloys" because their properties can be increased only by cold work. The 53-S alloy contains a constituent which makes possible a change in properties by heat treatment. As will be discussed in connection with brass, aluminum sheets of the various alloys are available in fully annealed, quarter-hard, half-hard, three-quarter-hard, and full-hard temper. In order to obtain similar properties in the finished part, design requiring more severe draws necessitates the use of more fully annealed sheets. Conversely, for shallow draws half-hard or even full-hard sheets may be used.

Providing the basis for an estimate of the number of operations required for the production of a given part, TABLE IV indicates the desired and permissible maximum reductions possible in a draw-

Fig. 12—Right—After drawing the material shown in Fig. 11, this rough surface results

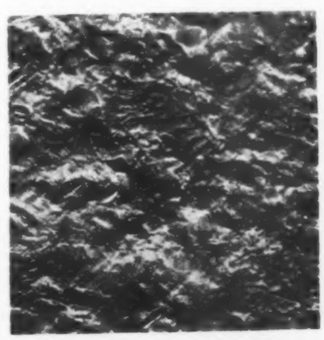


Fig. 13—Below—Stress-strain diagram for copper indicates the extent of increased power consumption of presses when drawing hard-temper material

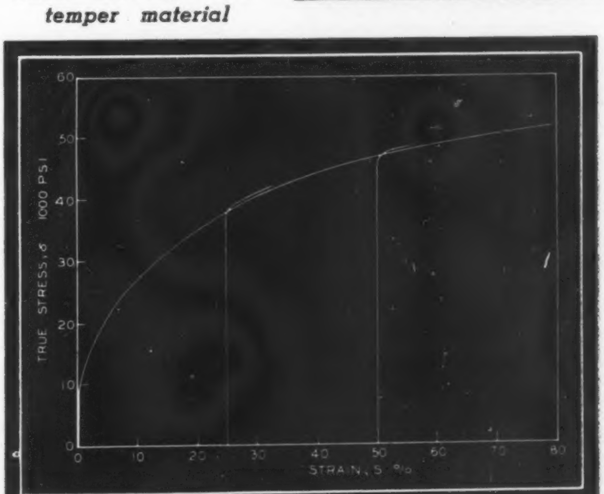


TABLE III

Mechanical Properties of Aluminum Alloy Sheet

Alloy and Temper	Alloy Strength (psi.)	Yield Strength* (psi.)	Elongation (% in 2 in., $\frac{1}{16}$-in. specimen)	Brinell Hardness (500-kg., 10-mm. ball)
2S-0	13,000	5,000	35	23
3S-0	16,000	6,000	30	28
52S-0	29,000	14,000	25	45
17S-0	26,000	10,000	20	45
17S-T	62,000	40,000	20	100
24S-0	26,000	10,000	20	42
24S-T	68,000	45,000	19	105
53S-0	16,000	7,000	25	26
53S-T	39,000	33,000	14	80

*Offset .2 per cent.

ing operation without intermediate anneals. As indicated, this table refers only to the 2S-0 and 3S-0 alloys, starting from the fully annealed condition. For harder tempers and other alloys such as 52-S the reductions per draw must be reduced. This decrease from the desired reduction may amount to as much as 10 per cent on the first draw and 5 per cent on succeeding draws depending on the hardness of the metal being fabricated.

Shown in *Figs.* 1, 2 and 3, are photomicrographs taken at 500 magnifications of the 2S-0, 3S-0 and 52-S sheets which are more commonly used for deep-drawing operations. The dark particles in *Fig.* 1 are the aluminum-iron-silicon constituent. Moderate grain size of this material together with the absence of any pronounced directionality of the grain boundaries makes it an ideal material for drawing. In *Fig.* 2 the fine and large particles are chiefly aluminum-manganese and aluminum-iron-manganese. In *Fig.* 3 the corresponding particles are largely aluminum-chromium-iron.

Providing a marked comparison with the lack of directionality evidenced by *Fig.* 1, the opposite condition is shown in *Figs.* 4 and 5 which are respectively photomicrographs of 24S-0 and 24S-T sheet. In the temper of the material shown in *Fig.* 5 the hardening constituents are retained in solid solution or exist as particles of submicroscopic precipitate.

A full cross section of .04-inch Alclad sheet is shown in *Fig.* 6. The core of this material is 24S-T alloy, while the surface coating layer is of high purity aluminum. In the boundary zone between the core and the coating, the diffusion zone resulting from the migration of copper and magnesium from the core to the surface layer is evidenced. At this magnification of 100 diameters a striking contrast is obtained between the direction of grain between the core and the lack of direction in the coating.

The character of the slip-lines developed by making a rockwell impression on the polished surface of a metallographic sample is illustrated by *Fig.* 7. This phenomenon is characteristic of aluminum alloy when stretched or drawn to failure and may be avoided by use of a more fully annealed alloy or by avoiding reduction per draw in excess of those indicated in TABLE IV.

Orange peel surface roughening resulting from a draw on a material with a relatively coarse grain is demonstrated by *Fig.* 8. The magnitude of such surface roughness will be realized from the fact that the magnification in this illustration is only twice actual size.

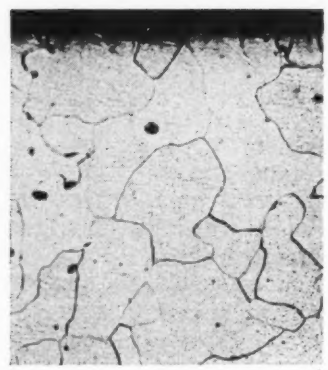


Fig. 15—Rimming steel similar to that in Fig. 14 has been heat treated to increase the grain size for higher ductility

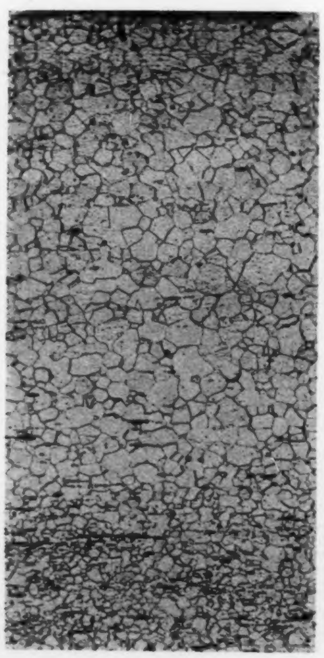


Fig. 14—Decrease in grain size toward the center of a sheet is characteristic of rimming steel as rolled or normalized. Magnification 100 diameters

Because of its wide range of obtainable properties brass is extensively used for drawn parts. It may be ordered in accordance with ASTM specification B36-41 which covers eight different brasses varying from a red brass containing 95 per cent copper and 5 per cent zinc to common yellow brass containing 65 per cent copper and 35 per cent zinc. The specification also covers five different anneals and seven different tempers although all alloys are not furnished in all anneals. The specification gives grain size and rockwell hardness value for the various anneals and tensile strength and rockwell hardness value for the rolled tempers.

For simple blanking and forming, these alloys are usually furnished in a temper one or two B & S numbers hard, which means that the sheet or strip was reduced in thickness one or two B & S numbers after the final anneal. The former is known as

"quarter-hard metal" or "quarter-hard temper" and the material two numbers hard is known as "half-hard" metal or temper.

Various anneals are used depending on whether medium draws are to be used to arrive at a finished article which must have a fine grain finish, or whether deep drawn cups are to be made with relatively few individual draws. For the latter purpose the anneal should produce a grain size of approximately .07 millimeters and it will be necessary to reanneal the cup after the area of the cross section has been reduced about 60 per cent. The surface can be restored to a fine grain finish by giving the metal a light anneal before the last drawing or forming. For deep draws, corresponding to such parts as cartridge cases, an alloy of 70 per cent copper and 30 per cent zinc or 72 per cent copper and 28 per cent zinc is usually used. Such alloys will take much deeper draw and can be worked more easily than alloys with a lower copper content. Furthermore,

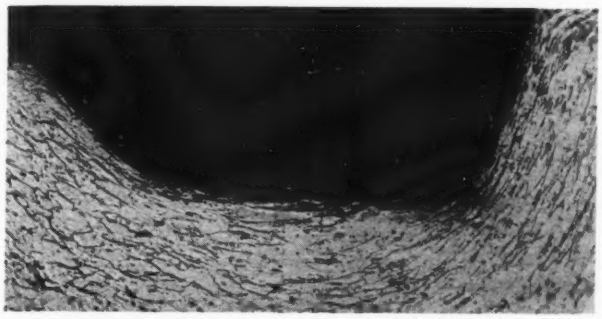


Fig. 16—Distortion and flow lines of the grain are caused by drawing operation

they will resist to a much higher degree the action of corrosion cracking.

An idea of the microstructure of deep-drawing cartridge brass may be obtained from *Fig.* 9 which shows the grain structure of this material at 75 magnification. *Fig.* 10 shows the same material at the same degree of magnification after the metal has been extremely hard worked. *Fig.* 11 shows the microstructure of a cartridge brass which produced an orange peel surface when drawn similar to that shown for aluminum in *Fig.* 8. This orange peel surface in brass is shown at eight magnifications in *Fig.* 12. Such a surface defect can be avoided by annealing the metal so that the grain size is less than .11 millimeters.

Usually brass with an elongation of 25 per cent in 2 inches or greater can be cold worked at room temperature. Material which has been thoroughly annealed and has a high elongation can be cold worked with about 60 per cent reduction in area of cross section. It will then be necessary to anneal the brass at not less than 950 degrees Fahr., pickle in sulphuric acid and clean before further cold working.

Impurities in brass, if too high, interfere with its good working properties. Iron and aluminum de-

Fig. 17—Coining or embossing operations on drawn parts cause grain discontinuity and crowding

crease the grain size and increase the tensile strength. Antimony and bismuth may cause difficulty in cold working.

Because of the constant reference to B & S numbers to indicate the amount of cold working to which a material has been subjected, it may be well to point out that the numbers are not selected on a purely arbitrary basis but have a definite relation to each other. For example an increase of six B & S numbers is tantamount to a 50 per cent reduction in thickness of sheet. When a material is said to be, for example, two B & S numbers hard, it means merely that in producing the material a reduction in thickness equivalent to two B & S numbers was performed in the rolls after the last annealing operation.

Whereas there is little if any difference in cost between fully annealed and full-hard sheets the latter, because of increased hardness and strength, are more difficult to draw. However, if the strength, hardness and rigidity of the completed part is critical it may be necessary to use sheets of a few B & S numbers hard in order to produce the desired physical characteristics of the finished part despite the fact that less expensive press operations could be obtained by using fully annealed sheets.

This consideration is portrayed graphically by *Fig.* 13 which shows the interdependence of tenacity and ductility of pure copper. As will be observed the stress required to perform a ten per cent strain on an annealed copper is around 27,000 pounds per square inch. In comparison, the stress required to effect a similar strain of ten per cent on copper

(*Continued on Page* 128)

TABLE IV

Allowable Reduction for 2S Aluminum

Draw Sequence		Desired Reduction (Per cent)	Maximum Permissible (Per cent)
Blank	D		
First Draw ...	D_1	40 D	42 D
Second Draw .	D_2	20 D_1	25 D_1
Third Draw ...	D_3	15 D_2	18 D_2
Fourth Draw ..	D_4	15 D_3	15 D_3

Simplifying Design of Ru

By C. W. Kosten and C. Zwikker

FACILITY with which steel helical tension and compression springs may be calculated and specified plus the ability of spring manufacturers to supply springs conforming to these specifications is responsible, in large measure, for the widespread use of special springs for specific applications. This has not been true in the case of rubber. Here, the admitted complications incident to precise and rigorous calculation have retarded the more extensive use of rubber spring elements.

Overcoming these handicaps to a large extent, this article (by means of simplified assumptions and by disregarding certain phenomena which within the scope of this work may ordinarily be considered negligible) enables the designer to handle problems involving the use of rubber in much the same manner as he would steel springs. In fact, the finally developed formulas resemble in form the conventional formulas for tension and compression springs. Serious attempts to achieve similar ends have been made, among others, by Ariano[1], Keys[2], Oeser[3] and Rocard[4].

In designing rubber springs it may safely be assumed that rubber can be adhesively vulcanized to metal. However, bearing in mind that rubber is vulcanized in molds, the shapes selected should be as simple as possible.

Throughout the subsequent development the physical concepts of the material are familiar, with the exception of the "modulus for 25 per cent stretch", *M*, which is used instead of Young's modulus. If it is recalled that Young's modulus is the "modulus for 100 per cent stretch" it will be evident that either may be used as a constant for a given material. Modulus for 25 per cent stretch, *M*, is defined as the stress in kilograms per square centimeter of the original cross-sectional area necessary to produce a deflection of 25 per cent of the length of the original unstressed piece. Without undue difficulty, rubber may be obtained with moduli (M) of 2, 4, 6, 8 and 10 kilograms per square centimeter.

RUBBER IN SHEAR: Modulus of rigidity, G, for rubber is defined as

$$G = \frac{\tau}{\tan \gamma} = 1.65\, M \qquad (1)$$

where τ denotes the shearing stress and γ the angle of shear. Since $\tan \gamma$ is proportional to τ this ratio is a constant even for large deflections of from 30 to 45 degrees. In some literature on the subject it is stated that G is a function of stress. This is usually owing to the application of a force at right angles to the direction of shear, exerted in order to increase the tolerated shearing stress. For example, in the conventional vibration absorbers shown schematically in *Fig.* 1, the rubber adjacent the vulcanized areas is restrained from axial displacement by the rubber-to-metal bond. Hence this bonding imposes, when torque is applied to the shaft, a load at right angles to the direction of application of shear.

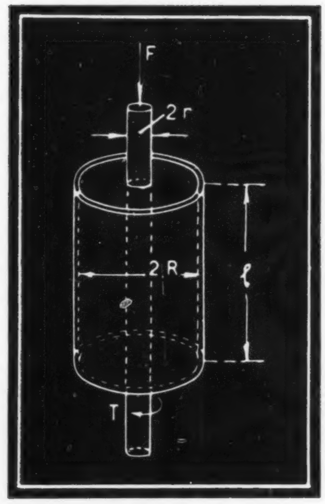
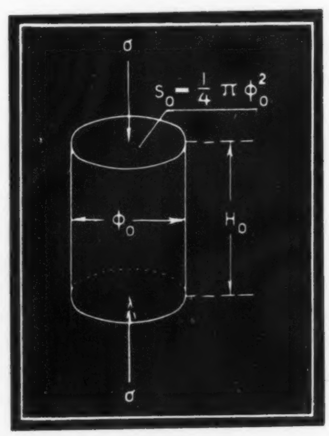


Fig. 1—Rubber cylinder bonded to two concentric cylindrical surfaces illustrates two directions of shear

Fig. 2--Below--Forces applied to ends of cylinder in tension or compression are effective only on a portion of height



NEED for a practical and simplified method of designing rubber springs has long been evident. The authors of the accompanying article have attained this end even though, in so doing, the ultimate in precision necessarily has been sacrificed. Attention to the stated limitations of the formulas, however, will result in rubber mountings fully satisfying most practical applications

Ru r Mountings

Apparent practical obstacles in the way of overcoming, in the design of the spring unit, this restraining effect necessitates a careful scrutiny of each application to determine the magnitude of probable variation in G. Reference to previous literature[5, 6, 7, 8] will prove enlightening.

RUBBER IN TENSION AND COMPRESSION: Since the formulas for the properties of rubber in tension are, taking cognizance of sign, applicable also to rubber in compression, the following analysis is developed with reference to the latter only.

[1] R. Ariano—*India Rubber Journal,* Vol. 76, Page 207, 1928.
[2] W. C. Keys—*Mechanical Engineering,* Page 345, 1937.
[3] K. Oeser—*Kautschuk,* Vol. 10, Page 121, 1934.
[4] Y. Rocard—*Journal de Physique et le Radium,* Vol. 8, Page 197, 1937.
[5] C. W. Kosten and C. Zwikker—*Physica,* Vol. 4, Page 221, 1937.
[6] C. W. Kosten—*Nederlandsch Tijdschrift voor Natuurkunde,* Vol. 4, Page 291, 1937.
[7] C. W. Kosten—*Procedures of the Rubber Technical Conference,* London, Page 987, 1938.
[8] C. W. Kosten—*De Ingenieur,* Vol. 54, Page 128, 1939. English translation as Communication No. 16 of the Rubber—Stichting.

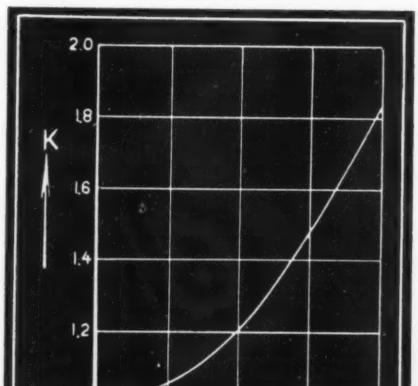


Fig. 3—Relative increase in stiffness under dynamic loading is a function of elastic modulus

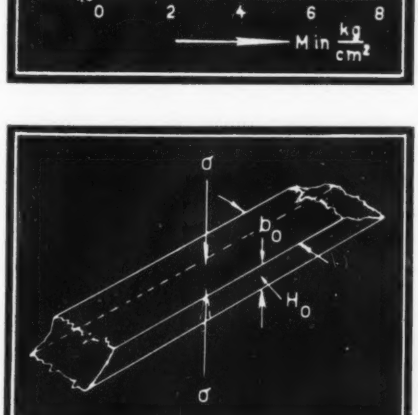


Fig. 4—As in Fig. 2, effective height in compression is a function only of the width of base of infinitely long rubber strip

TABLE I

Formulas for Static Deflection

(Referring to Figs. 2 and 4.)
Cylinders: E = 4.95 M
Strips: E = 6.60 M

Symbol	Formula	Approximation
σ	$\frac{1.07\,\epsilon\,E}{1.07-\epsilon}$	$\epsilon\,E$
ϵ	$\frac{\sigma}{E+.93\,\sigma}$	$\frac{\sigma}{E}$
s	$\frac{1.14\,S_0E}{h_0(1.07-\epsilon)^2}$	$\frac{S_0E}{h_0}$
G	$1.67\,M$	

TABLE II

Validity of Formulas in Table I

Quantity	Limits of validity for desired accuracy of 5 per cent	10 per cent
G	$-30° < \gamma < 30°$	$-45° < \gamma < 30$
ϵ	$-.90 < \epsilon < .60$	$-1.00 < \epsilon < .70$
s	$-.50 < \epsilon < .40$	$-.60 < \epsilon < .45$

TABLE III

Hardness Indices vs. Elastic Modulus

M (Kg/cm²)	Hardness: Shore Durometer Type A	Schoppers 10 mm ball	Pursey and Jones ¼" ball
2	31	150	160
4	44	98	107
6	55	68	74
8	63	53	57
10	68	43	47

Substantial incompressibility of rubber is the predominating factor controlling its properties. Therefore, Poisson's ratio for rubber is practically equal to 2. Also $G = E/3$, where E is Young's modulus. Further, all deformations occur at constant volume. Taking into account the constancy of volume, the relationship, $M = .2E$ is easily derived and is accurate to within a few per cent for 100 per cent stretch to 70 per cent compression.

When a cylinder of height H_0 and diameter ϕ_0, *Fig.* 2, is compressed to half its height the resulting shape, assuming no end constraint, will be a cylinder of height $H_0/2$ and diameter of $\phi_0\sqrt{2}$. As in steel springs, the stiffness of the cylinder after compression will exceed that before compression. Stiffness is defined as force per unit of deflection.

Another cylinder of the same kind of rubber having dimensions in its unstrained state of $H_0/2$ and $\phi_0\sqrt{2}$ possesses a stiffness equal to 4 times that of the first cylinder in its unstrained state. However, the first cylinder, compressed as indicated to the shape of the second, acquires a stiffness substantially equal to the second. The apparent stiffening of rubber by compression can, therefore, be attributed to the con-

Table IV

Compression of Rubber Cylinders

(Values of σ in kg/cm²)

ϵ	M (kg/cm²) 2	3	4	5	6	7	8	9	10
0.70 ..	19.9	29.8	39.8	49.7	60	70	79	89	99
0.60 ..	13.5	20.2	26.9	33.7	40.3	47.1	54	61	67
0.50 ..	9.3	13.9	18.5	23.2	27.8	32.4	37.0	41.7	46.3
0.40 ..	6.3	9.4	12.6	15.8	18.9	22.0	25.2	28.4	31.5
0.30 ..	4.11	6.2	8.2	10.3	12.3	14.4	16.4	18.5	20.6
0.20 ..	2.43	3.64	4.85	6.1	7.3	8.5	9.7	10.9	12.1
0.10 ..	1.09	1.63	2.18	2.72	3.26	3.81	4.35	4.90	5.4
0									
—0.20 ..	—1.66	—2.49	— 3.33	— 4.16	— 4.99	— 5.9	— 6.7	— 7.6	— 8.3
—0.40 ..	—2.87	—4.31	— 5.7	— 7.2	— 8.6	—10.1	—11.5	—12.9	—14.4
—0.60 ..	—3.79	—5.7	— 7.6	— 9.5	—11.4	—13.3	—15.2	—17.1	—19.0
—0.80 ..	—4.52	—6.8	— 9.0	—11.3	—13.5	—15.8	—18.0	—20.3	—22.6
—1.00 ..	—5.1	—7.7	—10.2	—12.7	—15.3	—17.8	—20.4	—23.0	—25.5

Table V

Compression of Rubber Strip

(Values of σ in kg/cm²)

ϵ	M (kg/cm²) 2	3	4	5	6	7	8	9	10
0.70 ..	26.5	39.8	53	66	79	93	105	119	132
0.60 ..	18.0	26.9	35.8	44.9	54	63	72	81	89
0.50 ..	12.4	18.5	24.7	30.9	37.0	43.2	49.3	56	62
0.40 ..	8.4	12.6	16.8	21.1	25.2	29.3	33.6	37.8	42.0
0.30 ..	5.5	8.2	10.9	13.7	16.4	19.2	21.8	24.7	27.5
0.20 ..	3.24	4.85	6.5	8.1	9.7	11.3	12.9	14.5	16.1
0.10 ..	1.45	2.18	2.91	3.62	4.35	5.1	5.8	6.5	7.2
0									
—0.20 ..	—2.21	— 3.33	— 4.44	— 5.5	— 6.7	— 7.9	— 8.9	—10.1	—11.1
—0.40 ..	—3.83	— 5.7	— 7.6	— 9.6	—11.5	—13.5	—15.3	—17.2	—19.2
—0.60 ..	—5.1	— 7.6	—10.1	—12.7	—15.2	—17.7	—20.2	—22.8	—25.3
—0.80 ..	—6.0	— 9.0	—12.0	—15.1	—18.0	—21.0	—24.0	—27.1	—30.1
—1.00 ..	—6.8	—10.2	—13.6	—16.9	—20.4	—23.7	—27.2	—30.6	—34.0

siderable deformations which are tolerated and which make such an increase in stiffness perceptible.

If no further complications existed, the behavior of rubber in tension or compression would be simple. However, as pointed out in conjunction with the discussion of shear, conditions incident to the application of load introduce constraining influences on its behavior. A cylinder subjected to axial compression will assume the shape of a barrel. This results from the fact that the end-compressing planes prevent the rubber in their immediate vicinity from extending sideways. Because the volume must remain unchanged, the outward extension is essential to compression. Hence the effective height of the cylinder as a spring is less than the total height. For cylinders the unelastic or ineffective height may be taken as $\phi/8$, thus being independent of height itself. This is, however, only a first approximation; limit of validity is the case where the two unelastic zones approach each other as in thin disks.

For most practical applications the correction factor, $\phi/8$, applied to the height will yield satisfactory accuracy so long as H_0/ϕ_0 exceeds ¼. For a cylinder 4 inches in diameter and 1-inch high, the effective height will be ½-inch. Also, in consequence of the fact that rubber parts in compression are usually made quite short the constancy of stress for each cross section and even for each point in the same cross section does not hold.

Dynamic and Static Stiffness Differ

A second complication stems from the difference between the dynamic and static stiffness of rubber springs. Mass effects in rubber are not given consideration here. These effects lead to characteristic frequencies as in the case of metal springs. For rubber these frequencies are higher and are, moreover, less dangerous because of internal friction[10].

Cyclic application of stress results in an increase in stiffness which varies with the frequency in accordance with the curve, *Fig.* 3. For soft rubber this amounts to only a few per cent and increases with hardness. In the curve, *Fig.* 3, K is the relative increase in stiffness and hence is the ratio of dynamic to static stiffness.

A second standard shape is the long strip of width b_0 and height H_0 as shown in *Fig.* 4. If a load is applied, compressing the strip in the direction of height, the characteristic lateral bulge occurs. Part of H_0 is, therefore, again unelastic, amounting to .22 b_0.

Owing to the length of the strip, the rubber possesses only a unilateral freedom of extension or bulge instead of the bilateral freedom of cylinders. Hence the stiffness is greater, equalling 4/3 that of the cylinder[9].

Referring to *Figs.* 2 and 4, Table I provides a convenient summary of the formulas based on the foregoing discussion. It will be observed that the values of the expressions for the several variables differ little from the simplified values in the right-

[9] Rocard's computations lead to the same factor 4/3. See reference 4.

[10] Internal friction is not dealt with in this article. Internal friction is expressed by the so-called angle of loss δ which is nearly a constant of the material, varying only slightly with frequency and shape. Such friction usually increases with increasing M and lies between 2 and 6 degrees. An undercured rubber has somewhat greater friction than a properly elastic one. For comparison, internal friction for spring steel is zero; for cork or synthetic rubber it amounts to about 5 degrees. Generally the conditions which the angle of loss must satisfy are of little consequence practically. Only an approach of this angle to zero would become dangerous and this is entirely impossible.

hand column. However, since the given values are only approximations it is essential, in using the table, that attention be given the limits of validity as controlled by the maximum allowable error as shown in TABLE II.

Since the property of hardness is one which, while being easily measured, is also usually used for rubber specification, TABLE III is of interest in that it provides data for determining M from a given hardness number.

In addition to plane shear given by formula (1), the case of rubber in shear between two concentric cylinders merits special attention. Referring to *Fig.* 1, when the force is applied as at F,

$$\text{Stiffness } (s) = \frac{2\pi\, lG}{\log_e (R/r)}$$

$$\text{Max. Shear } (\tan\gamma) = \frac{F}{2\pi\, rlG}$$

When a torque is applied as at T,

$$\begin{matrix}\text{Moment per radian}\\ \text{of displacement}\end{matrix} = \frac{4\pi\, lGr^2R^2}{R^2-r^2}$$

$$\begin{matrix}\text{Max. Shear at}\\ \text{inner cylinder}\end{matrix} (\tan\gamma) = \frac{T}{2\pi\, lr^2G}$$

In the case of rubber springs subject to dynamic load it is only necessary to multiply the M of the static formulas by the factor K obtained from the graph, *Fig.* 3, in which it is represented as a function of M. If a static load is imposed on the spring in addition to the dynamic load, the static deformation should be computed separately and added to the other.

Graphical solution for the static deformation of cylinders or strips under tension or compression may be obtained by means of *Fig.* 5. The deformation ratio, ϵ, is of course, computed by using the effective height in each case.

Similarly, the static stiffness, s, for cylinders and strips under tension or compression is found from the graph, *Fig.* 6. In this graph s_1 is the stiffness per unit of effective height, area and modulus and is related to s in accordance with the following expression

$$s = \frac{S_0 M}{h_0} s_1$$

The following two numerical examples demonstrate the use of the foregoing data in the calculation of rubber springs.

Examples Illustrate Use

EXAMPLE 1: Given a cylinder with $H_0 = 2.5$ centimeters, $\phi_0 = 5$ centimeters and a hardness of 45 Shore durometer units, find the amount of compression for a load of 50 kilograms and also the load necessary to produce a compression of .5 centimeters.

Solution: From TABLE III, $M = 4.2$ kilograms per square centimeter. From TABLE I, $E = 4.95$ $M = 4.95 \times 4.2 = 20.8$ kilograms per square centimeter. Also $\sigma = 50/(\frac{1}{4}\pi 5^2) = 2.54$ kilograms per square centimeter and, also from TABLE I, $E = .11$. Effective height, $h_0 = 2.5 - (\frac{1}{8})\, 5 = 1.87$ centimeters. Compression, therefore, equals $.11 \times 1.87 = .20$ centimeters.

For the second part, $E = .5/1.78 = .20$. Then from the formula for σ in TABLE I $\sigma = 7.5$ kilograms per square centimeter which is equivalent to a load of $7.5 \times (\pi/4) \times 5^2 = 147$ kilograms. Hence

TABLE VI

Initial Stiffness of Cylinders
(Kilograms)

$\varnothing_0$ (cm)	M (kg/cm²) 2	3	4	5	6	7	8	9	10
2.5 ..	49	73	97	122	146	170	195	219	243
5 ..	194	292	389	486	583	681	778	875	970
7.5 ..	438	657	877	1,095	1,310	1,530	1,750	1,970	2,190
10 ..	779	1,170	1,560	1,950	2,335	2,715	3,115	3,505	3,895
12.5 ..	1,220	1,825	2,435	3,045	3,650	4,260	4,870	5,480	6,090
15 ..	1,750	2,630	3,505	4,380	5,255	6,135	7,010	7,880	8,760
17.5 ..	2,385	3,580	4,770	5,970	7,160	8,350	9,550	10,740	11,930
20 ..	3,110	4,665	6,220	7,780	9,340	10,895	12,440	14,000	15,560
25 ..	4,870	7,300	9,740	12,170	14,600	17,040	19,480	21,900	24,300
30 ..	7,000	10,500	14,000	17,500	21,000	24,500	28,000	31,500	35,000

TABLE VII

Initial Stiffness of Strips
(Kilograms per square centimeter)

h_0/b_0	M (kg/cm²) 2	3	4	5	6	7	8	9	10
0.25	53	79	106	132	158	185	211	238	263
0.50	26.4	39.6	53	66	79	92	106	119	132
0.75	17.6	26.4	35.2	44.0	53	62	70	79	88
1.00	13.2	19.8	26.4	33.0	39.6	46.2	53	59	66
1.25	10.6	15.9	21.2	26.4	31.7	37.0	42.3	47.6	53
1.50	8.8	13.2	17.6	22.0	26.4	30.8	35.2	39.6	44
1.75	7.5	11.3	15.1	18.9	22.6	26.4	30.2	34.0	38
2.00	6.6	9.9	13.2	16.5	19.8	23.1	26.4	29.7	33

compression is .20 centimeters and load 147 kilograms.

EXAMPLE 2: Find the vertical resonance frequency of a motor weighing 1500 kilograms supported by four rubber cylinders of modulus $M = 4$ kilograms per square centimeter; $H_0 = 5$ centimeters, $\phi_0 = 10$. Also find the static compression.

Solution: Stress $\sigma = (1500/4) \times (\pi/4) \times 10^2 =$ 4.77 kilograms per square centimeter. Hence, from TABLE I, $E = .197$. Then $h_0 = 5 - \frac{1}{8} \times 10 = 3.75$ centimeters. Hence static compression $\epsilon h_0 = .74$ centimeters. Initial stiffness, obtained from TABLE VI, equals $1560/3.75 = 416$ kilograms per centimeter. Stiffness in dynamic loaded condition from TABLES VI and VII equals $4.6 \times 1.51 \times 1.20 = 754$ kilograms per centimeter. Resonance frequency in cycles per minute is then $(30/\pi)\sqrt{375000/754 \times 10^6}$ $= 428$ according to the frequency formula $(30/\pi)$ $\sqrt{s/m}$.

To facilitate calculation of rubber spring problems further, tables have been prepared which are similar in appearance and use to metal spring tables. Hence, TABLES IV and V give respectively the

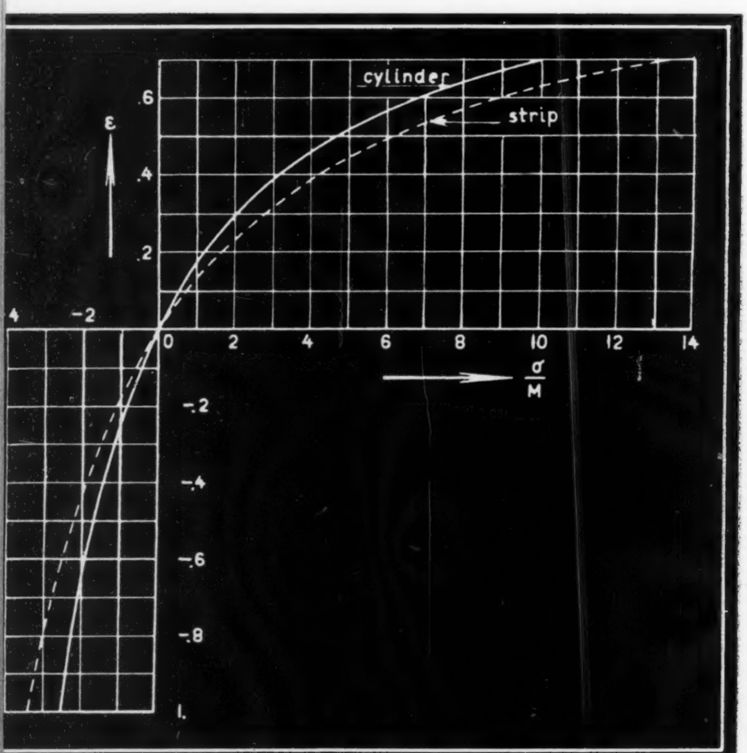


Fig. 5—Graphical solution of static deformation of cylinders and strips utilizes effective height

stress for cylinders and strips of rubber of modulus, *M*, necessary to produce a desired deflection ratio, ϵ.

Similarly, TABLES VI and VII afford, also for cylinders and strips respectively, easy determination of the stiffness in kilograms per centimeter of effective height in the unloaded condition. For stiffness in any loaded condition, the values obtained from TABLES VI and VII should be multiplied by the factor for stiffness ratio obtained from TABLE VIII. In the case of dynamic loading these values for stiffness should be multiplied, in addition, by the factor *K* given in TABLE IX.

In calculating rubber springs of shapes differing from cylinders and infinitely long strips, the following general rule stated by W. C. Keys[2] may be used: Rubber objects for which the ratio between the areas of the free (unloaded) and the non-free (loaded) surfaces are the same, will yield the same ϵ for the same σ. However, in comparing springs of too different shapes the accuracy provided by this rule is, at best, only a first approximation.

TABLE VIII

Static Change in Stiffness

(Cylinders and Strips)

ϵ	s loaded / s unloaded
.50	3.52
.40	2.55
.30	1.93
.20	1.51
.10	1.22
.00	1.00
—.10	0.84
—.20	0.71
—.30	0.61
—.40	0.53
—.50	0.46
—.60	0.41

TABLE IX

Dynamic Stiffness Increment

(Cylinders and Strips)

M (kg./cm²)	K
2	1.05
3	1.11
4	1.20
5	1.34
6	1.48
7	1.64
8	1.83

Greater accuracy may be obtained in the case of finite strips, rectangular blocks and elliptical cylinders by an extension of the foregoing analysis.

STRIPS OF FINITE LENGTH: Because of the somewhat greater sideways freedom, ϵ is larger than for an infinitely long strip. Compression causes an extension in length nearly equal to the increase in width. Relatively, the widening is ϵ and the elongation, therefore, is $(b_0/l_0)\epsilon$ where l_0 is the initial length. If thi selongation were pushed back by counted pressure in the length direction, half its amount would contribute to increased width and the other half would tend to decrease the compression, ϵ. Hence the analysis for the infinitely long strip can be applied, provided the computed value of ϵ is multiplied by the correction factor $1 + \frac{1}{2}\, b_0/l_0$. For $l_0 = 10\, b_0$ the correction therefore, amounts to only 5 per cent so that, for practical purposes, such strips may be considered infinitely long.

RECTANGULAR BLOCKS: In this case the approximation can be carried out in two ways, namely by a comparison with cylinders and with strips. Blocks with a relatively large length direction will resemble strips more closely than cylinders. The reverse, of course, is true for short blocks such as, for example, a cube. The approximation analagous to a strip is computed in a manner similar to the foregoing for finite length strips, i.e., strip formulas are applied; effective height is computed and the correction factor $1 + \frac{1}{2}\, b_0/l_0$ is applied to ϵ.

In the approximation analagous to a cylinder, the block is considered to be bilaterally free. Hence

cylinder formulas are applied. For determining the unelastic or ineffective part of the height, an arithmetical average of ¼ the width and ⅛ the length is taken. For short blocks, approaching the cube, an amount slightly larger than average may be used.

This method of computation will yield adequately accurate results only for comparatively small

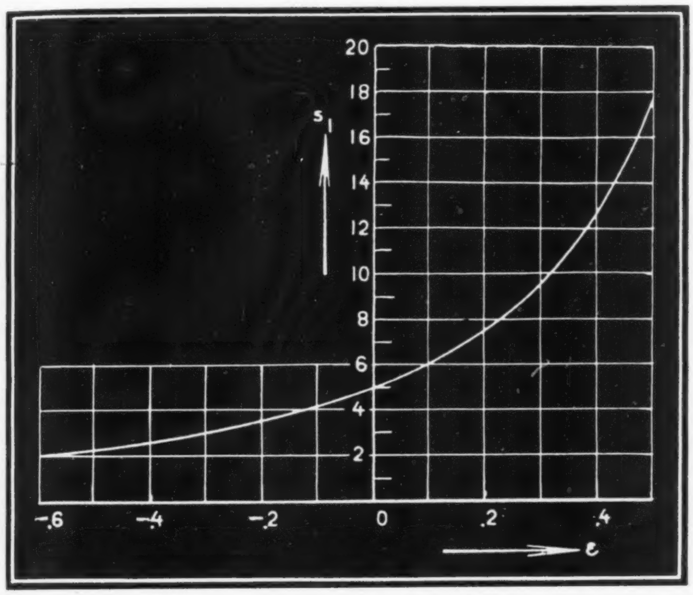


Fig. 6—Companion curve for Fig. 5 provides a solution for static stiffness of cylinders and strips

compressions. If large compressions are desired it is recommended that recourse be taken to the following procedure.

From a piece of properly elastic rubber ($M \approx 3$ kilograms per square centimeter) make one small test rod and three test blocks all having the same profile but differing in height. Determine the compression curves for these objects and write, for the selected profile, $h_0 = H_0 - ClB_0$ and $E = C_2M$ wherein B_0 is chosen consistently as either the longest or shortest dimension of the plane section perpendicular to the direction of loading. Constant C_1 and C_2 then remain to be determined.

Measure, by means of the test rod, the modulus for 25 per cent stretch, M. C_1 is then determined in such a way that the three compression curves coincide. From each set of values of σ and ϵ of this curve, C_2 is determined by means of the relation between σ, ϵ and E given in TABLE I. The two equations given are then valid for any objects conforming in cross section with the one investigated, regardless of height. All equations of TABLE I with the exception of those for E and h_0 apply equally well to any block, regardless of its profile.

Nomenclature

- M Modulus for 25 per cent stretch (kg/cm^2)
- σ Compressive stress in kg/cm^2 of original area
- ϵ Ratio of compression (H_0/h_0)
- H_0 Total initial height (cm)
- h_0 Effective height (cm)
- S_0 Cross-sectional area of unloaded piece (cm^2)
- E Young's modulus (kg/cm^2)
- τ Shear stress (kg/cm^2)
- γ Shear angle (radians)
- G Modulus of rigidity (kg/cm^2)
- s Stiffness (kg/cm)

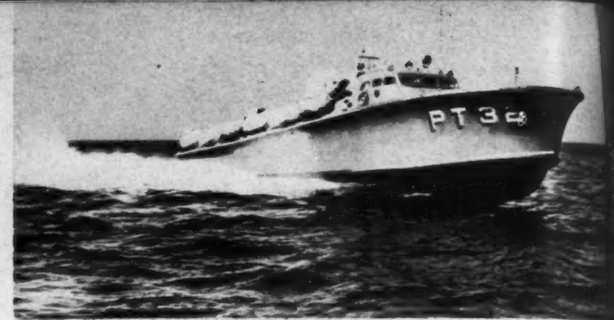
Substitutes Prove Superior

INDICATIVE of the resourcefulness of the engineer is his ability to develop and apply substitutes for priority metals. In many cases superior methods and materials have been found. In others, the substitutes—although more costly—are better or as good; in the remainder, they are "commercially satisfactory."

Typical are the substitutes shown in the accompanying table that are now being used or probably will be used by Westinghouse Electric & Mfg. Co. Quality and performance of each item are at least equal to accepted standards. Where an inferior but acceptable finish for instance, is used on one of the company's parts, the customer will be informed so that he may know what to expect. From the developments indicated in the foregoing and similar researches of other companies should come much of value, a tribute to the engineering profession.

Material	Substitute Material	Typical Applications
Nickel steel	Molybdenum steel	Shafts, bolts, gears
Tungsten steel	Molybdenum steel, small percentage tungsten	High speed tool steel
Nitriding steel	Chromium steel	Circuit breaker parts
18-4-1 Steel	High carbon, high chromium steel	Blanking dies
Stainless steel	High chrome steel	Electric range heater tubes
Aluminum	Copper	Disk for watt hour meter
	Brass	Shaft for watt hour meter
	Enameled steel	Electric roaster cover, electric range part, ice cube door
	Rubber, enameled steel sheets, plastics	Ice cube trays
	Rubber, brass, plastics, enameled sheet	Parts and agitator of laundry equipment
Aluminum die castings	Sheet steel with waterproof finish	Sockets for outdoor watt-hour meters
Zinc or aluminum die castings	Steel or brass	Handles and hinges for refrigerators
Brass	Steel plated with copper, brass or gilding metal	Bases for incandescent lamps
Nickel	Nickel plated steel or copper	Supports for incandescent lamps
Spun aluminum	Spun steel	Searchlight reflectors
Aluminum sheet alzak	Iron sheet with high reflecting porcelain enamel finish or glass with silvered finish	Reflectors
Zinc die casting	Cast iron	Fan bases
Silk	Cotton, glass, asbestos	Insulation fabric
Mulberry fiber	Old rope	Jap paper insulation
Tung oil	Dehydrated castor oil, soy bean oil, citicica oil	Insulation varnish

Douglas dive bomber is the largest plane ordinarily operating from the deck of an aircraft carrier. As a moden development, the ship is equipped with perforated wing flaps to reduce landing speed. Design of the plane was evolved from that of the BT-1 dive bomber and retains many of its features



Whereas detailed technical specifications of the PT 32 mosquito boat are not available, its predecesso the PT-10 costing $18,000, was 77 feet long, powere with three 1,225 horsepower V-12 Packard engine and has exceeded a speed of 50 knots

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Long-ranged navy bomber, Mars, built by the Gl Marti pable of flying non-stop to Europe and return, i wered carries the equivalent of a railroad tank car of fue ximum is 200 feet and overall length 117 feet 3 i Prop

Submarine, Thresher, of the Tambor class, completed in 1940, displaces 1,475 tons and is 299 feet in length. Utilizing an all-electric drive with General Motors diesels as the prime movers, the power plant develops 6,400 horsepower with which the ship was able to obtain 22 knots on its trial

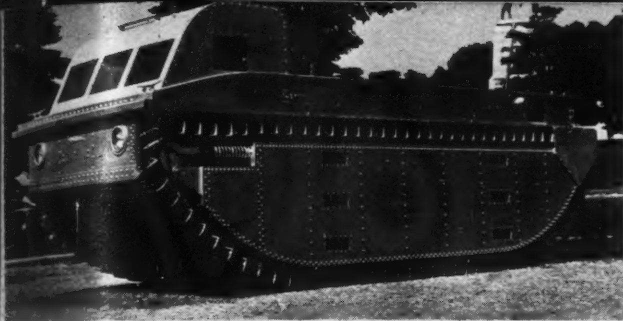
United States N

Featuring N l E

(Official U. S. N y Photog

Left—Powered w ared t aircraft carrier, H deve horsepower and $31,80 ship has a displ nt of and an overall le f 809 signed for a spee 4 kno was exceeded in l. An dition to numer chine eight 5-inch and 1.1-in craft guns, the an ac over anes

Right—Battleship, Texa ship of the New Y Having displacement of 3 ons, i in length overall. dition three planes it is d with guns, sixteen 5-in ns, a inch guns. Arm idship inches in thickne is red inches on either Gun t is 14-in d 8-inc



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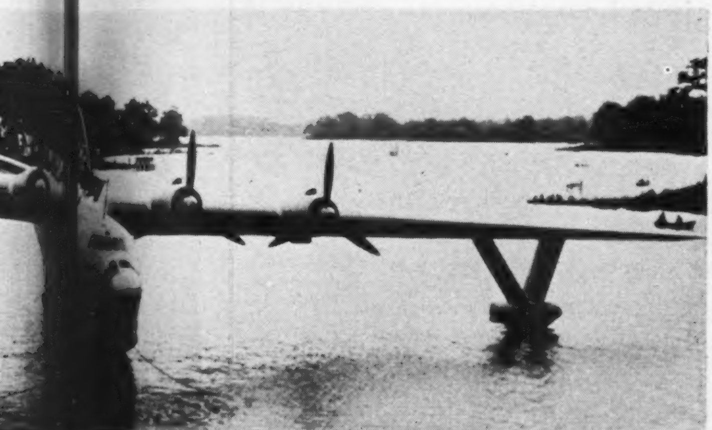
Amphibian tanks such as this have convincingly demonstrated their utility in extensive war maneuvers over difficult terrain. Equally mobile on dry land, in swamps, and in open water, these tanks would play an important part in repelling invasion

Grumman fighters of this type are in mass production for shipment to Britain as well as for the use of the United States Navy. All possible armor protection is provided for the pilot and the ship is fully equipped with rubber-lined, leakproof gasoline tanks and interconnecting piping



the Gl Martin Co. is the largest airplane in the world. Ca-
turn, it wered by four engines of 2,000 horsepower each, and
r of fue ximum normal weight is 140,000 pounds; wing span
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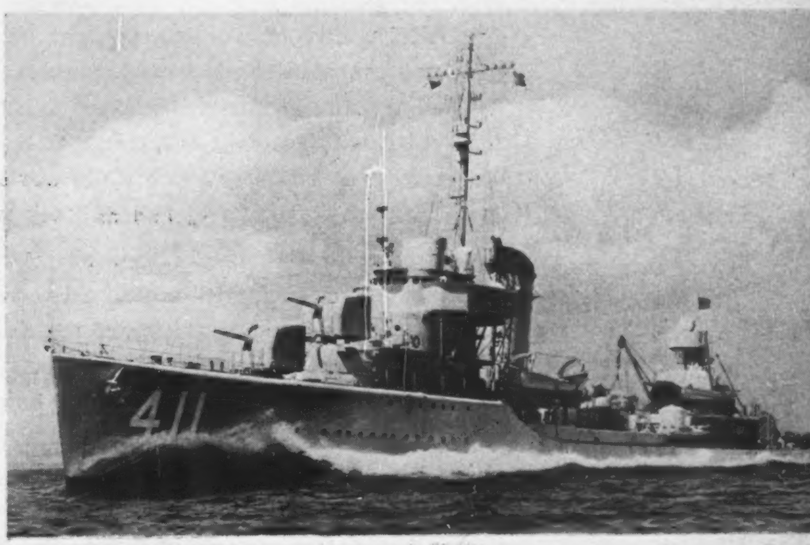

Destroyer, Anderson, completed in 1939, has a displacement of 1,570 tons and a speed of 36.5 knots. Using oil as fuel, the geared turbines develop 44,000 horsepower. Armed with five 5-inch guns and twelve 21-inch torpedo tubes the cost was $5,500,000

es War Machines

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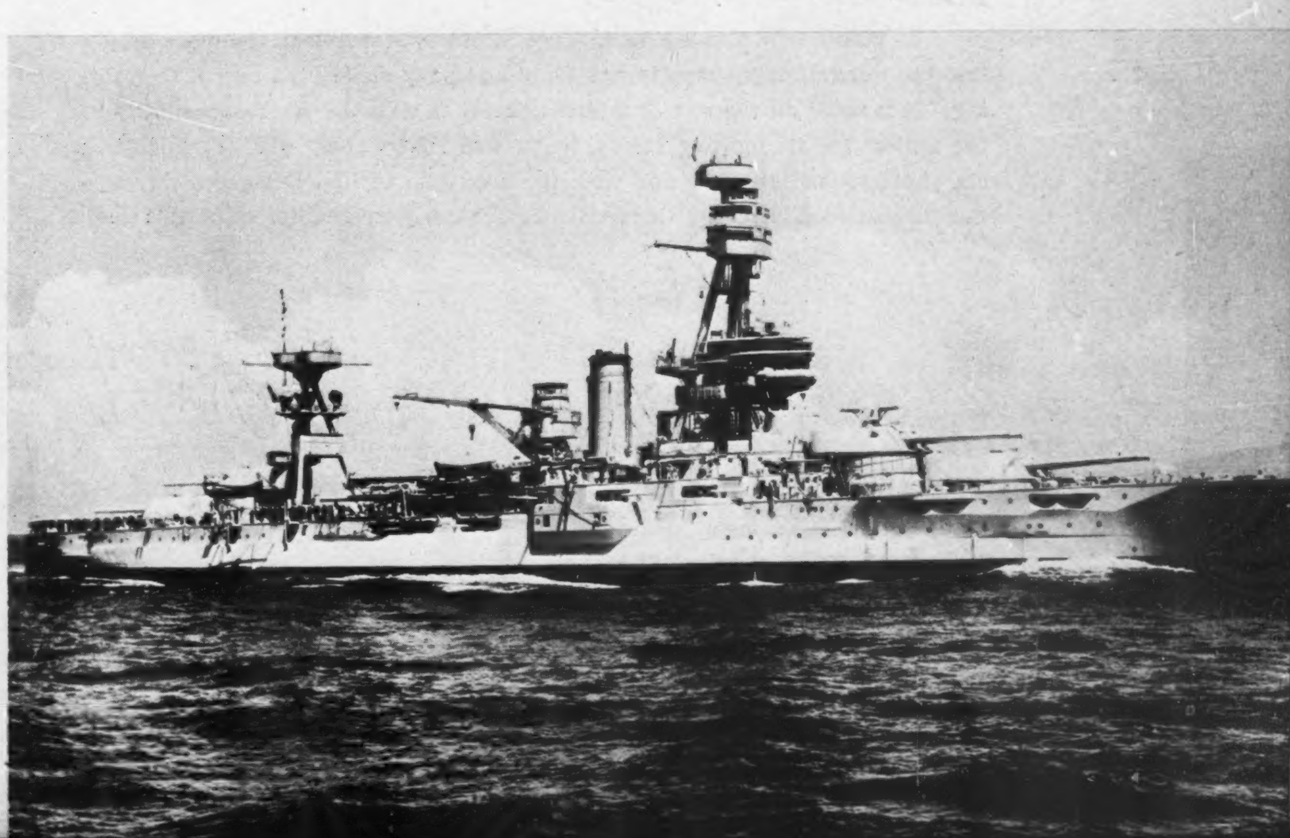

Ultimatum to Design!

STARK reality of war has brought into still sharper focus the role the American engineer is now destined to fulfill. To the sailor, the soldier and the airman the nation turns during these early days of actual conflict; but it will be to the engineer that the country increasingly will look as time goes on and the ups and down of war become more completely dependent on the superiority of naval and military equipment.

Faced with the development of effective war machinery—and of the machines for producing it—within a two-year period in comparison with the seven or more years preparation of the axis, United States designers already have done more than might be expected of them. That they will put even greater effort into their task in future, however, goes without saying in view of the spirit aroused by Pearl Harbor and the declarations of war against us.

It is to our fighting forces that eventually will come the honor of victory. Just when this will be achieved, however, rests to a large extent on the designers behind the men in the front lines. To underestimate the enemy's strength, courage or tenacity is to bluff ourselves into a false sense of security. But there is one thing above all others in which we can surpass them—the ability of our engineers to design and produce fighting equipment second to none.

The will to win, already his in large measure, is instilled even more deeply into the mind of every man in our army and navy during his course of training. It is up to all the rest of us individually to shoulder his responsibility to the nation and its fighting forces, to the end that victory will be reached in the shortest possible time and with the least loss of life. Designers throughout the nation will pledge themselves to the complete fulfillment of this goal.

L. E. Jermy

Design of Sections for Low Loads

By Hans A. Illing
Hydraulic Press Mfg. Co.

OBTAINING proper balance between desired rigidity and conservative use of material in light, low load-carrying machine parts is simplified by the following method which may be termed that of "equivalent stress". True, employment of conventional "beam theory" formulas will yield accurate results but this necessitates constant recourse to deflection formulas which, in addition to being often cumbersome, vary for different sections and systems of loading. Also machine designers accustomed to working with highly loaded members may have difficulty in visualizing the necessary rigidity of a part such as a lever which must support a load of only a few pounds. However, the design of a similar part to support a ton or more would present no complications.

This conception provides the basis for the "equivalent stress" method. The essence of the method is to estimate by experience a load, W_2, which if applied to a lever of the same length and stressed to a given working strength, S_2, would be sufficiently rigid. Knowing the load, W_1, to be applied to the actual lever, the stress, S_1, to be used to calculate the lever section can be obtained by substitution in

$$S_1 = S_2 \sqrt[4]{\frac{W_1}{W_2}} \qquad (1)$$

The relation is developed, beginning with an expression for section modulus, S_m,

$$S_m = C_1 \frac{W}{S} \qquad (2)$$

where W = load, S = allowable stress and C_1 = constant depending upon length, load distribution and kind of support.

If a square, circular or other simple section is to be used, the section modulus is proportional to the cube of the main direction, X, hence

$$S_m = C_1 \frac{W}{S} = C_2 X^3$$

and

$$X = C_3 \sqrt[3]{\frac{W}{S}} \qquad (3)$$

where C_2 = constant depending on the shape of the section.

The deflection, Y, of the beam is

$$Y = C_4 \frac{W}{I} \qquad (4)$$

where $I =$ moment of inertia of section and C_4 is a function of C_1 and the material used.

Moment of inertia may be found in a manner similar to the section modulus

$$I = C_5 X^4 \qquad (5)$$

where $C_5 =$ constant depending on the form of section.

Substituting (3) in (5)

$$I = C_6 \sqrt[3]{\left(\frac{W}{S}\right)^4}$$

and in (4)

$$Y = C_7 \sqrt[3]{\frac{S^4}{W}} \qquad (6)$$

Equation (6) represents the relation between deflection, working stress and load. It shows that deflection increases with decreasing load for beams calculated to constant stress. It also indicates that, under otherwise identical conditions, all beams having the same value of S^4/W will have identical deflections. Hence, if for two beams

$$\frac{S_1^4}{W_1} = \frac{S_2^4}{W_2}$$

the deflections will be the same. Equation (1) is derived directly from this relation.

For example, a cantilever 50 inches in length is to be designed with an end load (W_1) of 20 pounds using a square section of material capable of withstanding a working stress (S_2) of 10,000 pounds per square inch. By experience it is known that a lever of the same length designed to withstand a load (W_2) of 2000 pounds would be sufficiently rigid. The value of working stress to be used in design is then

$$S_1 = 10{,}000 \sqrt[4]{\frac{20}{2000}} = 3200 \text{ pounds per square inch}$$

Calculating the required section using this stress results in a bar 1¼-inch square. It will deflect about ⅛-inch. If the lever were calculated with the given working stress of 10,000 pounds per square inch, a bar 27/32-inch square would result which would deflect about ⅝-inch.

Some uncertainty, of course, is involved in the estimation of W_2. However, even a wide variation produces little change in dimension. Thus, if W_2 were taken as 20,000 pounds instead of 2000 pounds in the foregoing example, the resulting bar would have been increased only ¼-inch to 1½-inches square.

The same method of "equivalent stress" can also be applied to flat plates. The formula for such stress,

$$S_1 = S_2 \sqrt[3]{\frac{W_1}{W_2}} \qquad (7)$$

is deduced in a manner similar to that for beams or levers. In applying Equation 7 for flat plates, however, some difficulty may be experienced in selecting a "satisfactorily rigid analogy." In this case it may be desirable to determine upon an acceptable deflection and design to this specification.

ASSETS *to a* BOOKCASE

Belt Conveyors and Belt Elevators

By Frederick V. Heddel and Russell K. Albright; third edition, published by John Wiley & Sons Inc., New York; 439 *pages,* 6 *by* 9 *inches, cloth bound, available through* MACHINE DESIGN *for* $6.00 *postpaid.*

As an exhaustive and authoritative treatment of the subject implied by the title, this book cannot be too highly recommended. It is intended to be a practical book. It does not, however, contain descriptions of installations of conveying and elevating machinery, but aims rather to explain principles and the reason for doing things.

Standard present-day practices are discussed and compared, pointing out the advantages of each and why one compares favorably with the other in certain specific respects. Reasons for the obsolescence of some material handling equipment are minutely described.

The main section of the book is divided into two parts, one devoted to conveyors and the other to elevators, both of the belt type. In addition, several tables are appended giving the weights of materials handled by belt conveying and elevating equipment. According to the author's definition, the characteristic of an elevator which distinguishes it from a conveyor is that the former has, attached to it, buckets or other containers for receiving and moving the material.

The book is thoroughly illustrated with photographs, sectional and line drawings, showing in complete detail the several components of material handling equipment. As such it may be used by engineers and designers as a standard reference volume and guide for this type of machinery.

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Patent Fundamentals

By Leon H. Amdur; published by the Chemical Publishing Co., New York; 305 *pages,* 5½ *by* 8½ *inches, cloth bound, available through* MACHINE DESIGN, *for* $4.00 *postpaid.*

This elementary discussion of patent fundamentals assumes a minimum of initial knowledge on the part of the reader. As such it is a well written and interesting volume for use by engineers as a preliminary text or reference to more advanced work on this subject.

Importance of patents to engineers and industry cannot be overestimated. It is, therefore, highly desirable that engineers and designers have more than a speaking acquaintance with a subject which is so important to their interest. As an example of the method of treatment in this volume, the preparation and prosecution of an actual U. S. patent with reproduction of all papers and memoranda are fully set forth in one chapter. Full attention is also given to the definitions of what constitutes a patent, as it is to other considerations involving patentability, assignments, licenses, etc.

In this connection a number of patents which were refused by reason of lack of utility, immorality or frivolity, are discussed in some detail. As an example, an application claiming a novel type of mouse trap is of piquant interest. The inventor, undoubtedly an extremely humane person, claimed a mouse tray which instead of exterminating the rodent placed a collar, with a bell attached, around its neck. Based upon the theory that all mice are afraid of bells it was assumed that the mouse so equipped would scare all the other mice out of the house.

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Aerodynamics of the Airplane

By Clark V. Millikan; volume I, published by John Wiley & Sons, Inc., New York; 171 *pages,* 6 *by* 9 *inches, cloth bound, available through* MACHINE DESIGN *for* $2.50 *postpaid.*

At the request, in the spring of 1940, of the Lockheed Aircraft Corp. directed to the California Institute of Technology a course in aeronautical engineering was inaugurated for graduate non-aeronautical engineers. This book, the first of three volumes, is the result of a compilation of lecture notes delivered during the first part of this course. It contains much valuable information released for this purpose by the Lockheed Aircraft Corp. as well as the Douglas Aircraft Co. A valuable reference and text, it may be considered by designers and engineers as constituting the most advanced technical exposition of the subject currently available.

Completely detailed and analytical treatment of fundamental aerodynamic principles prefaces a similar coverage of basic data. This essential preliminary material is then applied to airplane per-

formance analyses and problems in longitudinal and lateral stability.

The two supplementary volumes of this series will shortly be forthcoming. Complete series, all of which will be based on the graduate engineering course, is known as the Galcit series. Following the tendency to alphabetize the unwieldly names of bureaus, each letter in the name is the initial of each of the words in the Guggenheim Aeronautical Laboratory, California Institute of Technology.

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Engineering Kinematics

By Alvin Sloane, Assistant Professor of Mechanical Engineering, Massachusetts Institute of Technology; published by the Macmillan Co.; 310 pages, 6 by 9 inches, cloth bound; available through MACHINE DESIGN *for $4.00 postpaid.*

Departing from what regrettably has come to be a conventional presentation of the subject of kinematics, this book is refreshing by reason of its stress on fundamentals. Displacement, velocity and acceleration are the main headings to which the subordinate subjects of linkages, cams, gears, etc., are related.

The difference in approach will be appreciated immediately. Instead of taking, for example, a linkage system and analyzing it from the standpoint of the three considerations of motion each of these considerations is detailed separately with reference to the same linkage. The engineer is thus impressed by the simple fact that displacement, velocity and acceleration constitute both the foundation and superstructure of the subject of kinematics.

As has long been customary in the handling of this subject, the kinematic analysis of motion is predicated upon specific mechanisms. However, as is well known to designers, it is often essential that a mechanism be designed to provide a certain motion. Of course, when a mechanism is to be used to provide a given motion, which mechanism is mathematically analogous to one of conventional design, no difficulty is experienced. Solution of problems which constitute exceptions to these cases is, however, difficult to carry out. This condition is universal in kinematic texts and is one to be deplored.

Plane Engine Drafting Standards

Impetus given to the aircraft industry as a result of the present armament program has served to emphasize the necessity for standardization of airplane engine drafting room practice. This work has just been completed by a committee working under the Aircraft-Engine Subdivision of the Aeronautics Division of the S.A.E. Standards Committee and is available in a 37-page loose-leaf manual for $1.50 from the society's headquarters in New York.

The committee consisted of representatives of the Pratt & Whitney Aircraft Corp., Lycoming Div. of the Aviation Mfg. Corp., Packard Motor Car Co., Allison Div. of the General Motors Corp., Wright Aeronautical Corp. and the Ranger Aircraft Engines Co. Included in the manual are all the essential data on the preparation of drawings, arrangement of views, lines and line work, sectional views, dimensioning, screw thread representation, etc. Standards for dimensioning by the decimal system are also stated.

Because of the extensive amount of subcontracting work being done for airplane engine manufacturers, the necessity has long been felt for a universal system which would facilitate the filing of prints and drawings. This need has been recognized by the committee as is evidenced by standards providing for the uniformity of drawing sizes, accordion folding of blueprints, arrangement and location of number blocks, etc. General notes for concentricity, parallelism, circularity and case hardening have been systematized along with standard definitions and decimal designation for material instead of the former use of gages, names and numbers.

Substitute Coatings

SUBSTITUTES for galvanized steels are discussed fully in a new pamphlet published by the American Iron and Steel Institute. Entitled "Possible Substitutes for Zinc Coatings on Steel" the booklet recognizes that replacing zinc coatings which have been developed over the years and found satisfactory involves problems and probable cost increases in addition to occasional sacrifice in quality.

Recommendations are included for various substitutes as well as preparing the steel surface for coating. The discussion is divided into four groups, consisting of (1) Lead base coatings, (2) Metals unavailable or of restricted availability, 3) Nonmetallic inorganic coatings, and (4) Organic coatings.

Lead is covered at length because it is the most promising alternate to zinc. It is a flexible coating, adheres to surface, is an excellent surface for painting, and acts as a lubricant for stamping operations. Disadvantages such as electronegative action which accelerates corrosion and pin holes which start corrosive action are recognized. Various methods for processing lead coatings are compared.

Aluminum, chromium, copper, zinc, cadmium, nickel and tin are treated as metals of restricted availability with the recommendations that use of any be scrutinized carefully. Nonmetallic inorganic coatings include cement, oxides, phosphates, vitreous enamel and "zinc and phosphate" coatings. Surface treatments preparatory to coating are divided into light metallic, chemical and nontreatment.

AVAILABLE FOR PROMPT DELIVERY

With production schedules constantly endangered by non-delivery of essential materials, the elimination of even one such problem is a valuable aid in maintaining maximum output. Torrington Needle Bearings are available for ***prompt*** shipment on your priority orders!

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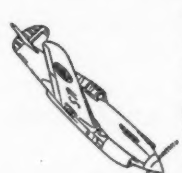
REDUCTION IN WEIGHT



and availability when needed are among the outstanding reasons for the use of Needle Bearings in many aircraft applications. A typical instance is its use on the cowl flap control rod of P-40 pursuit planes built by Curtiss-Wright Corporation.

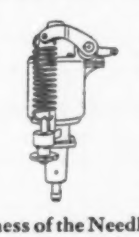
CURTISS—WRIGHT

COMPLETE RELIABILITY

is essential in such materials handling equipment as the Towmotor Lift Truck, which is subjected to severe service and heavy loads. Towmotor assures reliable bearing performance by using Needle Bearings on steering axle and hydraulic lifting cylinder crosshead.

TOWMOTOR LIFT TRUCK

HIGH CAPACITY IN SMALL SPACE



is obtained by the use of Torrington Needle Bearings on the rocker shaft of Class 2600 Governors built by The Pickering Governor Company. Compactness of the Needle Bearing contributes to simplification in product design and to accurate governor operation.

PICKERING GOVERNOR

formance analyses and problems in longitudinal and lateral stability.

The two supplementary volumes of this series will shortly be forthcoming. Complete series, all of which will be based on the graduate engineering course, is known as the Galcit series. Following the tendency to alphabetize the unwieldly names of bureaus, each letter in the name is the initial of each of the words in the Guggenheim Aeronautical Laboratory, California Institute of Technology.

□ □ □

Engineering Kinematics

By Alvin Sloane, Assistant Professor of Mechanical Engineering, Massachusetts Institute of Technology; published by the Macmillan Co.; 310 pages, 6 by 9 inches, cloth bound; available through MACHINE DESIGN *for $4.00 postpaid.*

Departing from what regrettably has come to be a conventional presentation of the subject of kinematics, this book is refreshing by reason of its stress on fundamentals. Displacement, velocity and acceleration are the main headings to which the subordinate subjects of linkages, cams, gears, etc., are related.

The difference in approach will be appreciated immediately. Instead of taking, for example, a linkage system and analyzing it from the standpoint of the three considerations of motion each of these considerations is detailed separately with reference to the same linkage. The engineer is thus impressed by the simple fact that displacement, velocity and acceleration constitute both the foundation and superstructure of the subject of kinematics.

As has long been customary in the handling of this subject, the kinematic analysis of motion is predicated upon specific mechanisms. However, as is well known to designers, it is often essential that a mechanism be designed to provide a certain motion. Of course, when a mechanism is to be used to provide a given motion, which mechanism is mathematically analogous to one of conventional design, no difficulty is experienced. Solution of problems which constitute exceptions to these cases is, however, difficult to carry out. This condition is universal in kinematic texts and is one to be deplored.

Plane Engine Drafting Standards

Impetus given to the aircraft industry as a result of the present armament program has served to emphasize the necessity for standardization of airplane engine drafting room practice. This work has just been completed by a committee working under the Aircraft-Engine Subdivision of the Aeronautics Division of the S.A.E. Standards Committee and is available in a 37-page loose-leaf manual for $1.50 from the society's headquarters in New York.

The committee consisted of representatives of the Pratt & Whitney Aircraft Corp., Lycoming Div. of the Aviation Mfg. Corp., Packard Motor Car Co., Allison Div. of the General Motors Corp., Wright Aeronautical Corp. and the Ranger Aircraft Engines Co. Included in the manual are all the essential data on the preparation of drawings, arrangement of views, lines and line work, sectional views, dimensioning, screw thread representation, etc. Standards for dimensioning by the decimal system are also stated.

Because of the extensive amount of subcontracting work being done for airplane engine manufacturers, the necessity has long been felt for a universal system which would facilitate the filing of prints and drawings. This need has been recognized by the committee as is evidenced by standards providing for the uniformity of drawing sizes, accordion folding of blueprints, arrangement and location of number blocks, etc. General notes for concentricity, parallelism, circularity and case hardening have been systematized along with standard definitions and decimal designation for material instead of the former use of gages, names and numbers.

Substitute Coatings

SUBSTITUTES for galvanized steels are discussed fully in a new pamphlet published by the American Iron and Steel Institute. Entitled "Possible Substitutes for Zinc Coatings on Steel" the booklet recognizes that replacing zinc coatings which have been developed over the years and found satisfactory involves problems and probable cost increases in addition to occasional sacrifice in quality.

Recommendations are included for various substitutes as well as preparing the steel surface for coating. The discussion is divided into four groups, consisting of (1) Lead base coatings, (2) Metals unavailable or of restricted availability, 3) Nonmetallic inorganic coatings, and (4) Organic coatings.

Lead is covered at length because it is the most promising alternate to zinc. It is a flexible coating, adheres to surface, is an excellent surface for painting, and acts as a lubricant for stamping operations. Disadvantages such as electronegative action which accelerates corrosion and pin holes which start corrosive action are recognized. Various methods for processing lead coatings are compared.

Aluminum, chromium, copper, zinc, cadmium, nickel and tin are treated as metals of restricted availability with the recommendations that use of any be scrutinized carefully. Nonmetallic inorganic coatings include cement, oxides, phosphates, vitreous enamel and "zinc and phosphate" coatings. Surface treatments preparatory to coating are divided into light metallic, chemical and nontreatment.

AVAILABLE FOR **PROMPT** DELIVERY

With production schedules constantly endangered by non-delivery of essential materials, the elimination of even one such problem is a valuable aid in maintaining maximum output. Torrington Needle Bearings are available for *prompt* shipment on your priority orders!

Right from the start, steadily rising sales curves reflected industry's immediate recognition of the Needle Bearing's advantages. Long before defense requirements brought increasing demands for most industrial materials, Torrington was prompted to expand production capacity again and again, to anticipate growing use of the Needle Bearing. So broad was the scope of this expansion program, initiated in the light of the Bearing's demonstrated potentialities in normal times and its acceptance by so many of the country's leading engineers, that it has proved sufficiently flexible to care for the abnormal requirements of the present.

Manufacturers of aircraft, automotive equipment, textile machinery, farm implements, electrical apparatus, and many other products have put the Needle Bearing to work in thousands of applications, because of such conspicuous features as low cost, small size, high capacity, ease of installation, efficient lubrication. And today the Needle Bearing has a new and even more important appeal—its ready availability to the makers of defense equipment and machines.

You can avoid delays by incorporating the Needle Bearing in your designs *now*—and you can profit as well by its adaptability to the solution of a wide range of bearing problems. Torrington engineers will be glad to work with you in incorporating the advantages of the Needle Bearing in your product. For full details on this unusual bearing, write for Catalog No. 109.

THE TORRINGTON COMPANY
TORRINGTON, CONN., U. S. A. • *Estab. 1866*
Makers of Needle and Ball Bearings
New York Boston Philadelphia Detroit
Cleveland Chicago Los Angeles Seattle
San Francisco Toronto London, England

REDUCTION IN WEIGHT

and availability when needed are among the outstanding reasons for the use of Needle Bearings in many aircraft applications. A typical instance is its use on the cowl flap control rod of P-40 pursuit planes built by Curtiss-Wright Corporation.

CURTISS—WRIGHT

COMPLETE RELIABILITY

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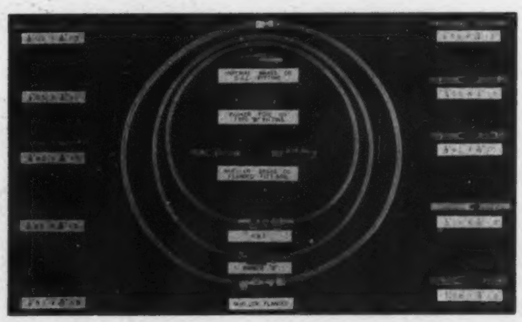
PICKERING GOVERNOR

New PARTS AND MATERIALS

Flexible, Plastic Tubing

PLASTIC tubing, made of Saran plastic, is now being fabricated by The Dow Chemical Co., Midland, Mich., into a flexible, semi-transparent tubing which can be used for oil lines for motors having central oil systems, gasoline lines for automobiles, tubing for recording devices and gages, refrigerant transfer, electrical insulation, humidifier supply lines, etc. It is claimed that Saran because of its toughness has already displaced strategic materials such as copper, nickel, stainless



steel and ceramics in several fields. The tubing is now available in sizes from ⅛-inch to ⅝-inch outside diameter, with various wall thicknesses of 0.30-inch to .062-inch. Saran is a thermoplastic having excellent resistance to water and brines, straight chain alcohols, ethers, ether-alcohols, alcohol-esters, and aliphatic hydrocarbons. Tests of tubing having 5/16-inch outside diameter and a .062-inch wall, joined with couplings, showed that it withstood a pressure of 1500 pounds per square inch. In a fatigue test, tubing was flexed through an angle of 15 degrees, 1750 times per minute, for 2,500,000 cycles without failure.

Hard-Faced Solenoids Announced

TO ELIMINATE difficulty of plunger "mushrooming", the solenoid recently announced by The National Acme Co., 170 East 131st street, Cleveland, uses welding in its construction. This welded protection consists of cutting narrow transverse grooves in three surfaces of the plunger and one in the armature, and by welded stellite deposit,

No. 2 of a series designed to help Industry prepare for the day when "Defense Production Stops"

"But... WHEN ~~DEFENSE~~ War PRODUCTION STOPS?"

"Will my company be ready with new products, new equipment, new designs to compete in the coming scramble for business?"

It isn't too soon to think about that, now that the defense load is shifting from "design engineering" to "mass production". When defense production stops, it will stop suddenly. It takes time to develop new ideas.

Though our increased manufacturing facilities are now devoted all-out to meeting vital defense needs, we want to help those companies for the future.

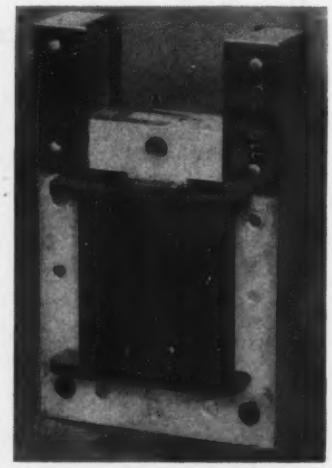
To such companies the facilities of our engineering department are wide open: To design the necessary Cone-Drive gearing for you, without charge or obligation.

You will find that Cone-Drive gearing will improve your designs and cut your costs with its greater load carrying capacity, greater compactness and longer life.

We will be glad to send you a bulletin on "Why Cone-Drive Gearing". #CW-41A (For executives); CW-41 (For design engineers).

CONE-DRIVE DIVISION MICHIGAN TOOL COMPANY
7171 E. McNichols • Detroit, U.S.A.

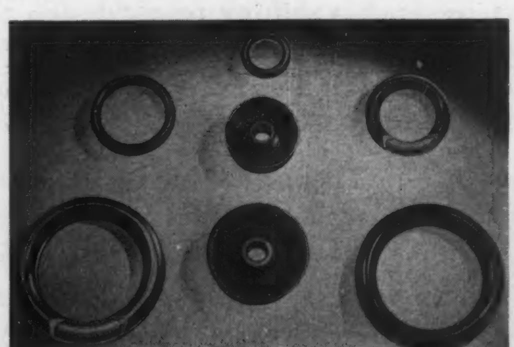
effecting hardened surfaces on both armature and plunger at their vital point of impact. The welding in the plunger arms eliminates residual friction and prevents wearing grooves in laminations and misalignment with the guide plates. The weld deposit is nonmagnetic. Hand-wound coils on steel spools have heavy bakelite ends. Coil insulation consists of three layers of tough varnished fabric, finally covered by a fully-impregnated and varnished cord. The solenoids are available in sizes with ratings from 4 to 30 pounds, push or pull, at 1-inch stroke. Combination push-pull and other special applications can be furnished. Pounds pull



indicated are over and above plunger weight. These controls are furnished for 110-220-440-550 volts and 25-40-50-60 cycle. The solenoids are available both for constant or intermittent duty. For constant duty the coil may remain energized as long as the plunger is completely seated; for intermittent, where the coil will not be continuously energized more than three minutes at one time, or where the "off" time is greater than the "on" time.

Synthetic Rubber for Sealing

AN ANNOUNCEMENT of importance to the aviation industry covers the new material known as Ameripol, manufactured by Miller Rubber Co. Inc., Akron, O. The formula for this synthetic rubber is an exclusive one with the Miller company. It is suitable for oil seals, V-type packings, accumu-



THE STORY OUR ADVERTISEMENTS DO NOT TELL

The advertisements of The New Jersey Zinc Company which are appearing currently in these pages explain the reasons why non-defense consumers have not been able to obtain all of the zinc they would like to have.

At the close of a critical business year, however, The New Jersey Zinc Company feels that it should tell a story which would not normally be a part of such a series of advertisements—a story of the cooperation given to this Company by its customers.

This cooperation, and the resulting exchange of vital information, made possible:

A better understanding of our mutual problems and difficulties;

The elimination of unreasonable demands and unfilled promises;

The fulfillment of contract obligations in spite of a tremendous expansion of business;

The shipment of a large list of zinc products where and when they were needed most—in defense and non-defense fields.

Consequently, thanks to our customers' cooperation and understanding, we feel that a difficult task has been accomplished.

The New Jersey Zinc Company looks forward to 1942 with confidence that this cooperation—which has proved of the greatest value under a heavy strain—will continue to operate to the mutual advantage of all concerned.

160 FRONT STREET

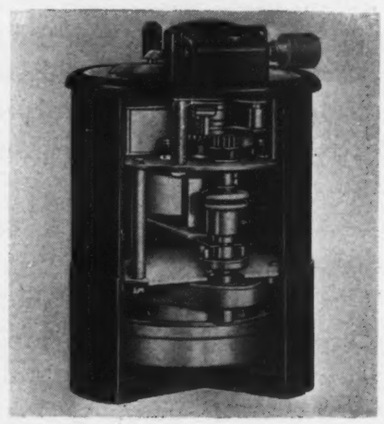
New Jersey zinc

NEW YORK, N. Y.

lator diaphragms, and for sealing all parts on airplane hydraulic systems. A particular advantage of the material is its minus 50 deg. F. resistance to sub-zero temperatures. Low oil absorption, low friction, ability to be fabricated to close tolerances, and high resistance to metal corrosion, heat, light and ozone are other characteristics. The accompanying illustrations show typical oil seals and packings which have already been tested and approved by aviation authorities and are now in production.

Industrial Time Delay Relay

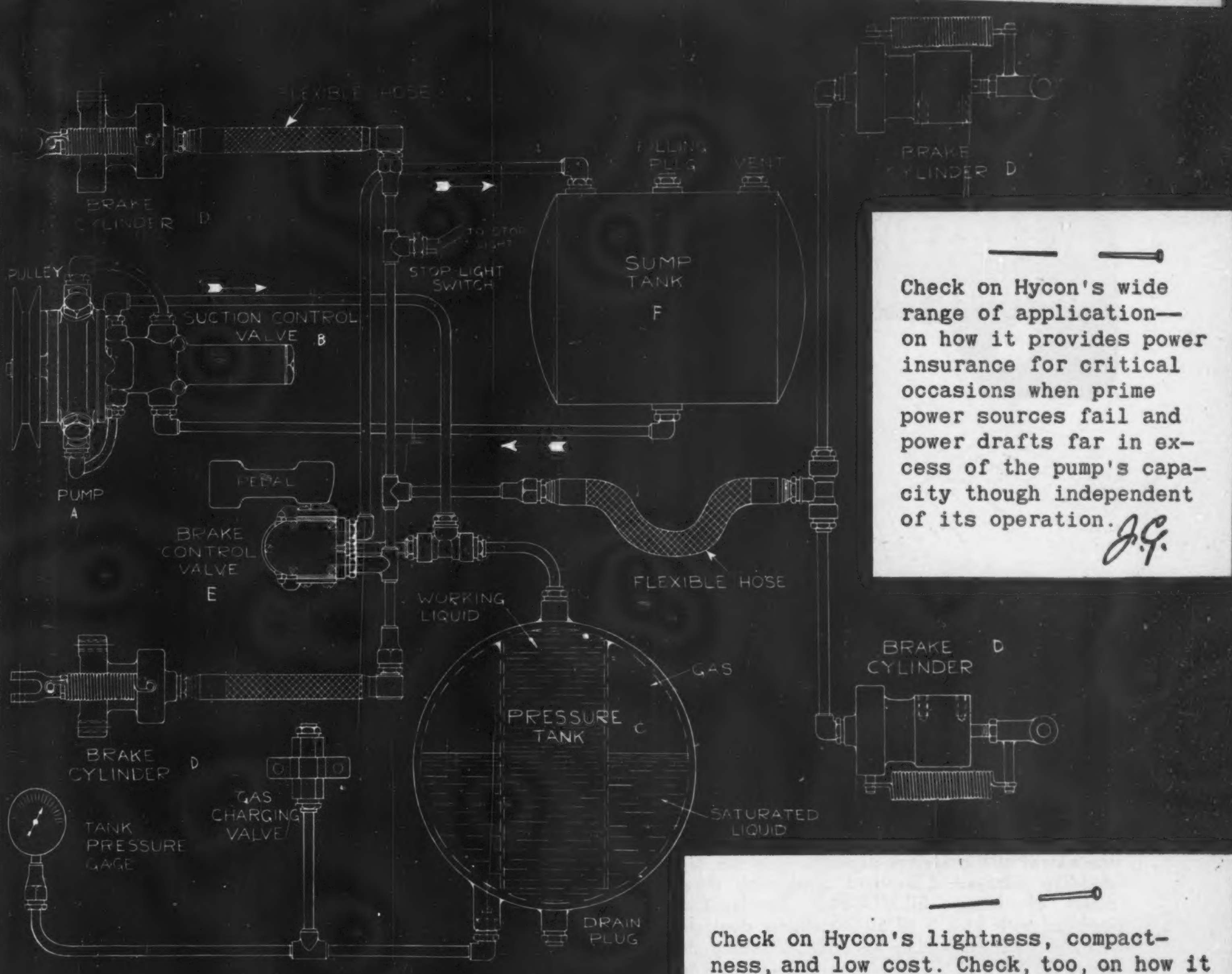
TIME delay relays, announced by The R. W. Cramer Co. Inc., Centerbrook, Conn., have an instantaneous reset feature, and when restarted repeat the timing without resetting. Developed particularly for industrial application, these compact relays consist of a synchronous motor unit which rotates a switch operating arm when magnetic clutch mechanism is energized. When clutch magnet de-energizes, it releases switch operating



arm, returning it to a starting position by means of reset spring. Starting position of switch arm determines time setting and is controlled by micrometer setting knob, located on top of housing, or a fractionally held stop arm, if concealed adjustment means is desired. Main switch unit is quick-make, quick-break, with silver contacts rated 10 amperes, 115 volts or 5 amperes 220 volts, or 2 amperes 440 volts alternating current. A ½-horsepower motor load can be operated satisfactorily, a 1000-watt heater load, or 250-watt lamp load. This slow-speed synchronous motor develops 450 revolutions per minute and is of self-starting type, with dust-tight enclosed gear train.

Metal Has High Tensile Strength

BABBITT metal for bearings subject to high presures and temperatures has been developed by Magnolia Metal Co., 120 Bayway, Elizabeth, N. J. With a tensile strength of 17,500 pounds per square inch, a yield point of 6500 pounds per square inch



Check on Hycon's wide range of application— on how it provides power insurance for critical occasions when prime power sources fail and power drafts far in excess of the pump's capacity though independent of its operation. J.G.

Check on Hycon's lightness, compactness, and low cost. Check, too, on how it saves power by eliminating constant pumping on intermittent operations and operates under all ordinary temperature conditions without adjustment. J.G.

HOW THE HYCON SYSTEM ACTUATES AND CONTROLS SPEED AND PRESSURE IN STRAIGHT LINE MOTION MECHANISMS WITH STORED HYDRAULIC PRESSURES UP TO 3000 LB PER SQ IN.

Typical installation with Hycon pump and pressure tank actuating 4 work cylinders

he diagram shows a typical assembly of HYCON equipment. The ork cylinders (D) may serve any desired function such as the moveent of a brake, clutch lever, door, electric circuit breaker. To these, draulic pressure is applied in any amount up to the designated mit by the control valve (E). The energy for actuating the work cylinders is primarily derived from the pressure tank (C) and this in turn is charged by the pump (A). The delivery from the pump is governed by the suction control valve (B). The work cylinders on the return stroke discharge into the sump tank (F) from which the fluid is pumped into the pressure tank and/or the work line.

GET THE ACTS ON HYCON

are facing the problem of designing a control or control actuatstem for any mechanism employing straight line motion, inforn on Hycon is important to you. With it you can achieve greater city of design, accuracy of control, and, probably, increased ne capacity at lower cost. Details concerning the system and rking parts are covered in the recently prepared booklet, lopments in Systems for Hydraulic Control of Straight Line n." Please let us know where to send your copy.

HYDRAULIC CONTROLS, INC. ***(Division of The New York Air Brake Company)***
122 South Michigan Avenue, Chicago, Ill.

Hydraulic Controls, Inc., Dept. 106
122 South Michigan Avenue, Chicago, Ill.
Please send me my copy of "Recent Developments in Systems for Hydraulic Control of Straight Line Motion" to:

Name..................................

Company..................................

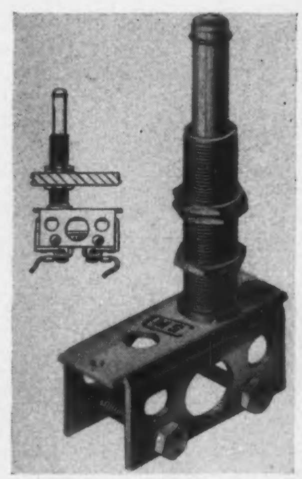
Street Address..................................

City......................*State*..........

and a brinell hardness of 27, the metal has a pouring temperature ranging from 950 to 1000 degrees Fahr. It is resistant to extreme local heat due to its high softening and melting temperatures and its strength makes it adaptable to heavy bearing loads such as are encountered in railroad service, heavy rolling mill machinery and paper mill machinery. Nickel treatment gives it a hard glossy surface, desirable for generators, motors and other high-speed applications.

Panel Mounting Actuator Bracket

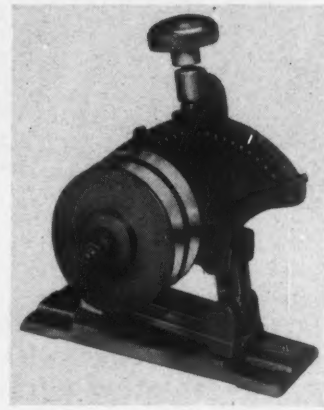
IN THE form of a bracket, designed for panel mounting, the Micro Switch Corp., Freeport, Ill., has just brought out its new switch actuator. Primarily for aircraft use, it also fits in many other applications because of its sturdiness of assembly, protection of switch unit, the ease with which switch unit can be replaced without removing actuator from its panel mounting, and the fact that the location of the point of operation of switch can be changed through adjusting panel mounting. Mounting is provided by three hexagon nuts on the panel mounting bushing. Two of these are used to position switch on panel or strut on which it is mounted, and the third locks assembly in place. Actuator can be mounted in holes 15/32-inch in diameter on panels up to 1⅜-inch thick. Different lengths of plungers are available. Type M-2 allows for ¼-inch overtravel and has a short bushing; Type M-7 allows for ⅞-inch overtravel and has a long bushing; and Type M-27 is a hybrid having ¼-inch overtravel and a long bushing.



Variable Pulley Control

OPERATING on the conventional variable pulley principle, the JFS-Cub variable speed transmission introduced by Standard Transmission Equipment Co., 416 West Eighth street, Los Angeles, is designed for all "A" section V-belt applications and for speed ranges up to 3.3-1. Smooth-sided pulleys are used rather than interlocking type, rotating on special bronze bearings which are provided with forced lubrication at all times. Speed changes may be made by shifting pulley spindle toward or away from the motor and driven machine, thereby automatically changing the driving ratio of variable pulleys. This can be done while

machine is in motion in two or three seconds by merely shifting control lever and locking in place by a turn of the knob. The positive belt align-



ment feature of this control makes possible the mounting of the transmission in any position without impairing function or throwing belts out of alignment. Free-end pulley spindle permits easy installation of belts.

Explosion-Proof Switch Offered

DUST-TIGHT, explosion-proof switches introduced by Appleton Electric Co., 1758 Wellington avenue, Chicago, are available as oil-immersed or air break units. Momentary pushbutton switches are operated by threaded shaft with rocker handle which is protected by guards for locking in any desired position. Bottom of switch is equipped with a dome-shaped cover or tub which, when filled with oil to level mark, will immerse switch mechanism in oil. Threaded cover on top provides easy access. Making of connections is simplified by a 4-wire block. Vertical through-feed hubs are provided on the body for threaded conduit. Removable close-up plug permits either dead-end or through-feed wiring. The switch is available with single or double operating handles, for normally open or normally closed circuit, or any combination

Welded Stainless Tubing

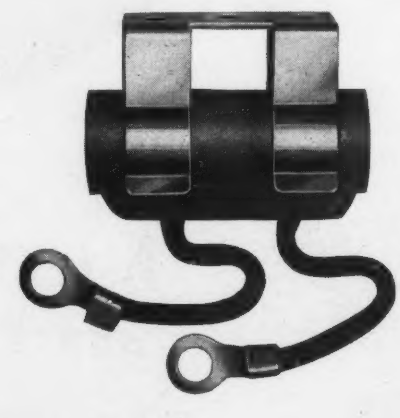
DEVELOPED after considerable experiment with various welding processes, a stainless steel tubing produced by a closely-controlled electric welding process is now ready for production at Globe Steel Tubes Co., Milwaukee, Wis. This new welded stainless steel tubing provides a high degree of corrosion resistance, strength with mini-


MINERAL OIL. Most tracing papers are treated with some kind of oil. Mineral oil is physically unstable, tends to "drift", never dries completely. Papers treated with mineral oil pick up dust, lose transparency with age.
VEGETABLE OIL, chemically unstable, oxidizes easily. Papers treated with vegetable oil become rancid and brittle, turn yellow and opaque with age.
ALBANITE is a crystal-clear synthetic solid, free from oil and wax, physically and chemically inert. Because of this new stabilized transparentizing agent Albanene is unaffected by harsh climates—will not oxidize with age, become brittle or lose transparency.
A crystal-clear synthetic solid age-proofs this tracing paper
No oil, no wax—but a remarkable new transparentizing agent developed in the K&E laboratories—produces this truly permanent tracing paper! ALBANENE is made of 100% long fiber pure white rags—treated with *ALBANITE*—a new crystal-clear synthetic solid, physically and chemically inert. ALBANENE will not oxidize, become brittle or lose transparency with age.
Equally important, ALBANENE has a fine hard "tooth" that takes ink or pencil beautifully and erases with ease ...a high degree of transparency that makes tracing simple, produces strong sharp blueprints... extra strength to stand up under constant corrections, filing and rough handling. ALBANENE has *all* the working qualities you've always wanted—and *it will retain all these characteristics indefinitely.*
Try ALBANENE yourself on your own drawing board. Ask your K&E dealer, or write us, for an illustrated brochure and a generous working sample.
EST. 1867
KEUFFEL & ESSER CO.
NEW YORK—HOBOKEN, N. J.
CHICAGO · ST. LOUIS
SAN FRANCISCO · LOS ANGELES · DETROIT · MONTREAL
K&E Albanene
REG. U. S. PAT. OFF.
THE STABILIZED TRACING PAPER

mum weight, and uniformity of structure. The method of welding is such as to result in a tube with very little "flash", thereby reducing to a minimum amount of grinding or cutting off of flash required to obtain a smooth finished tube. Readily bent, coiled, swaged and formed, the tubing is used in chemical and process industries, food industries, pulp and paper, oil and other industries where high corrosion resistance is a factor. To be marketed under the tradename "Gloweld" the tubing already has been successfully adopted in aircraft construction for hydraulic lines, structural and engine parts, where corrosion resistance is needed coupled with light weight and strength. It will be made available in a wide range of diameters and wall sizes, and in practically all the stainless steel analyses.

Mercury Switch Introduced

SIMPLICITY is foremost in the design of the new mercury switch announced by Bacon Electric Timer Corp., 4513 Brooklyn avenue, Cleveland. Of single or double-throw type, the switch has features of a glass mercury switch, but will not break.



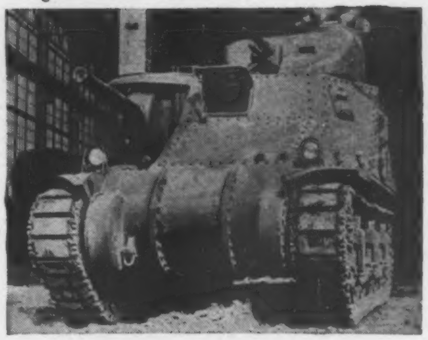
It comprises a metallic cartridge in two parts, sealed together and pressed into a tube of insulating material, then sealed again with lead wires (any length) soldered direct to the metallic cartridge which is filled with inert gas under pressure. Unit is available in sizes for any requirements.

Springs Unaffected by Temperature

NOW available commercially, a new spring, announced by All-Weather Springs, 72 Washington street, New York, utilizes the property of a 36 per cent nickel steel to become stiffer as its temperature increases. Such springs are combined with undercorrecting springs; that is, springs having an opposite tendency or with other elastic elements such as bellows, diaphragms, etc., to produce instruments, the accuracy of which is virtually unaffected by temperature changes. While ordinary steel or alloy springs become approximately 2 per cent more resilient for every increase of 100 degrees Fahr. in temperature, the new self-compensating springs can

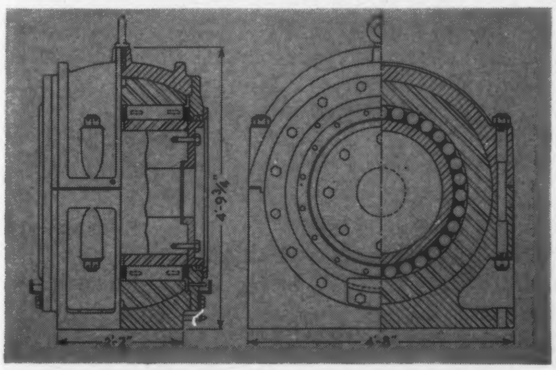
IN THE NEWS

WITH BANTAM BEARINGS

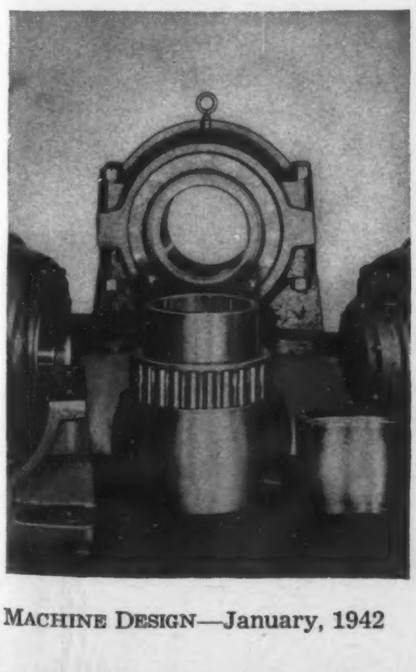


"BIG BRUISER" is this M-3 28-ton tank shown leaving the assembly line of the Chrysler Tank Arsenal. In the transmission, Bantam's Quill Roller Bearings aid in passing the tremendous driving power from motor to tractor treads. Maintaining a prompt, efficient flow of bearings of all types to the assembly lines of industry is Bantam's first job in helping meet the demands of the rapidly increasing production of defense material.

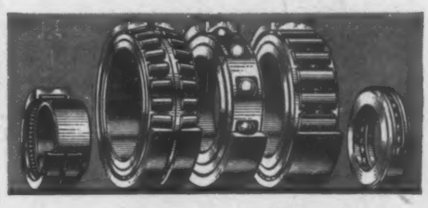
ENGINEERING ADVANCEMENT for new Chicago project is the use of radial roller bearings in the bascule type bridge now under construction for North State Street. The bridge, a perspective study of which is shown above, will rank among the largest of its kind in the world, with a width of 108 feet and a clear span of 210 feet. Two 36-foot roadways will be provided. Shown at left is cross-section of one of the large (47½" O.D.) Self-Aligning Roller Bearings, designed and built by Bantam, which will support two sets of three trunnions which in turn will carry the two movable leaves weighing 8,500,000 lbs. each. Self-aligning features, provided by spherical outer races resting in spherical housing seats, will compensate for any possible load deflections that may take place—another example of Bantam's engineering skill in designing anti-friction bearings for new and unusual requirements. Furthermore...



Substantial savings in size of electrical equipment and in power operating costs will be made through the use of Bantam Radial Roller Bearings. Only four 75 HP series motors will be required to operate the bridge, while the lower servicing and maintenance requirements of these bearings will bring additional savings. Bantam Bearings, such as those shown at left, have already been applied with gratifying results in other types of bridges. Breakaway and running loads have been reduced, and spans are moved with greater ease.



THE MANY ADVANTAGES of Bantam's compact, high capacity Quill Bearing are making it increasingly in demand for defense needs, where dependability and long-time maintenance-free service are primary considerations. A self-retained unit, easily installed and lubricated, it is ideally adapted to production line assembly methods. And its small size permits savings in materials of surrounding parts. For details on this unusual bearing, write for Bulletin B-104



EVERY MAJOR TYPE OF ANTI-FRICTION BEARING is included in Bantam's line—straight roller, tapered roller, needle, and ball. If you have a difficult bearing problem, TURN TO BANTAM.

BANTAM BEARINGS

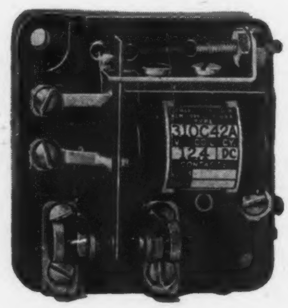
STRAIGHT ROLLER · TAPERED ROLLER · NEEDLE · BALL

BANTAM BEARINGS CORPORATION · SOUTH BEND · INDIANA

be held constant within these limits to .02 per cent. The springs are recommended for use in spring scales, aeronautical instruments and other spring-actuated instruments which are exposed to a wide variation in temperature.

Highly Sensitive Power Relay

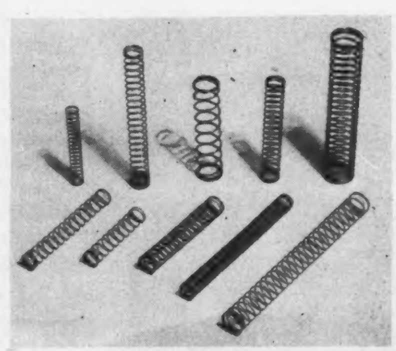
MOUNTED on a molded-bakelite base, a new sensitive power relay has been introduced by Kurman Electric Co., 241 Lafayette street, New York. This relay operates on .0018 watts, and is accurate within 2 per cent, with an output of 200-500 watts. The means provided for adjusting air



gap and spring tension in connection with the steep curve of the magnetic material used, allow adjustments for exacting requirements. Oversize coil may carry 3 watts above rating without overheating. Available in series 21 and 22, the relay operates with voltmeter at temperatures varying from minus 20 to plus 90 degrees Cent.

Beryllium Copper Springs

BERYLLIUM-COPPER brush springs developed by Instrument Specialties Co. Inc., 242 Bergen Boulevard, Little Falls, N. J., have high safe operating temperatures and are supplied to the following tolerances: Length, 1/32-inch, outside diameter

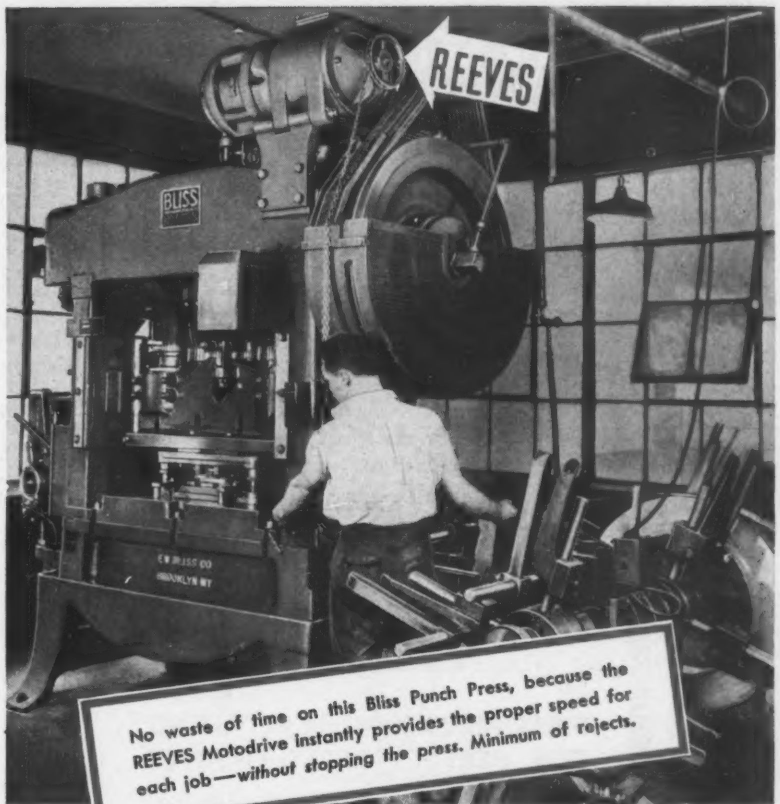


.003-inch. Load deflection is 5 per cent. Uniform in pitch, the springs can be compressed to within ⅛-inch of solid length without coils touching. Friction against brush-holder is reduced as the springs are sufficiently straight to roll easily across a flat surface. High safe working stresses permit springs

Yesterday

When Rigid Economy Was Paramount . . .

When competition was at its keenest and getting costs down and profits up was the order of the day in every plant, REEVES Variable Speed Control became firmly established in industrial production. Then, as now, REEVES Speed Control made it possible to improve both the quality and quantity of output and to save costly man-hours of labor by enabling the operator to run his machine at *exactly the right speeds* for best results on every job.

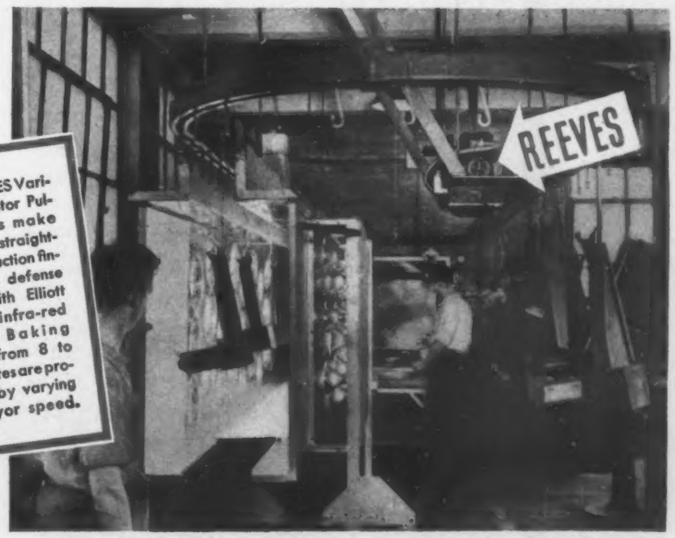


No waste of time on this Bliss Punch Press, because the REEVES Motodrive instantly provides the proper speed for each job—*without stopping the press.* Minimum of rejects.

Today

When Production Must Hit a New Peak . . .

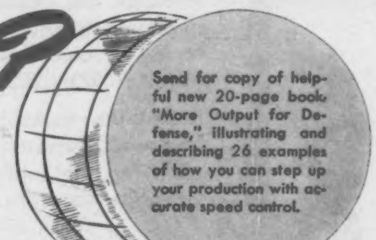
Today there is no time to waste with inefficient methods, with machines operating too fast or too slow for best results. Every man-hour of labor must be made to count. Every yard, pound or gallon of precious materials must be conserved. Production must hit a new peak, but with no sacrifice of quality. REEVES Drives, which provide complete speed flexibility, are a basic factor in accomplishing these objectives.



Here REEVES Vari-Speed Motor Pulley helps make possible straight-line production finishing of defense parts with Elliott Electric infra-red oven. Baking cycles from 8 to 18 minutes are provided by varying conveyor speed.

-and Tomorrow?

Tomorrow your machines may be called upon to handle entirely different materials, produced under entirely different conditions. *Applied now,* REEVES Speed Control will pay dividends in increased machine efficiency, *after the war* as well as in the present emergency. Let our engineering department help you.



Send for copy of helpful new 20-page book, "More Output for Defense," illustrating and describing 26 examples of how you can step up your production with accurate speed control.

REEVES PULLEY COMPANY • *Dept. H* • COLUMBUS, INDIANA

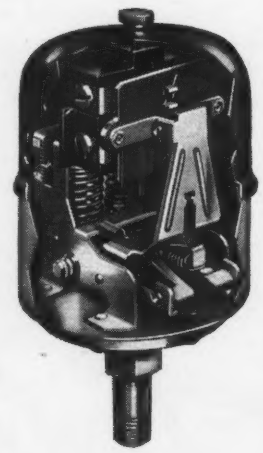
Minimum Waste - Maximum Production

REEVES Speed Control

to be redesigned for greater free length, reducing the change in brush pressure and wear. Being heat-treated for minimum drift, the springs maintain initial assembled tension without set. Special heat-treating results in high conductivity which eliminates necessity for shunts or pigtails.

Electric Pump Switch

FOR electric pump control, Sampsel Time Control Inc., Spring Valley, Ill., has made available its new automatic pressure switch. This switch, designed to control pressure in water systems where an electric pump is used to maintain a given pressure, and for controlling stoker systems, is operated by means of a rubber diaphragm moving a pivot disk. This disk actuates a free-floating toggle link mechanism thereby causing both sides of the line to open or close instantly. A waterproof cover eliminates possibility of water entering the switch mechanism. The switch will operate single or polyphase alternating current motors up to 2 horsepower on 110 volts or 3 horsepower on 220 volts, and a ½ horsepower or less on 32, 110, or 220 volts direct current. It can be supplied either with or without a short pipe nipple for tapped hole mounting, fitting a ¼-inch standard pipe thread, and mounted in a radius of 1¾ inches.



Small Air-Powered Pump

ADDITION of a new model midget size air powered pump is announced by Eastern Engineering Co., 45 Fox street, New Haven, Conn. The air-powered motor of this Model "D" pump makes it positively explosion-proof. It has variable performance, controlled by varying the speed of the vane type air motor. The pump can be used in all industrial, pilot plant, laboratory and experimental applications where an explosion-proof unit is required and compressed air is available. Size of the pump is 5½ x 6 x 7 inches, and weight is 10 pounds. Air consumption is 5 cubic feet per minute, maximum speed 4000 revolutions per minute on 100 pounds maximum air pressure. The pump can be furnished in stainless steel, monel metal, chromium plated bronze, and other alloys.

Electrolytic Blackening Process

ELECTROLYTIC blackening almost all metals including aluminum, steel, zinc, cadmium, nickel, tin and various alloys, a process of The Enthone

Consistent with the Demand of Modern Machine Tools . . . this Clutch has

4 Extra Advantages

● Several years ago, in cooperation with a few prominent machine tool builders, the Twin Disc Clutch Company designed and built this single hub, duplex MT Clutch. It has been standard equipment on some prominent machine tools for the past two years.

Now that exhaustive tests have been completed . . . the correctness of the design fully established . . . and the necessary changes in production made—these clutches are available to all manufacturers of machine tools in these sizes: 3″, 3½″, 4″, 4½″, 5″, 5½″, 6″, 7″, and 8″.

Advantages of the Twin Disc Single Hub, MT Clutch

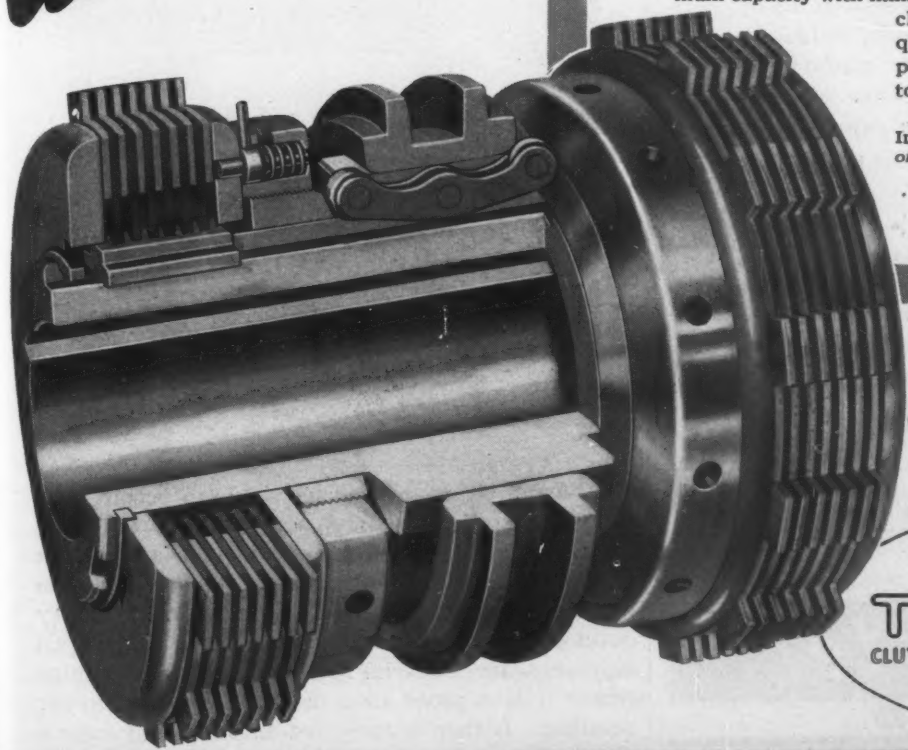
1. The pivoted, single, operating lever assures positive disengagement of the opposite clutch when either clutch is engaged. Likewise, the positive disengagement of both clutches when in neutral position.

2. In the single hub design there are fewer pins, therefore, less points subject to wear.

3. The use of widespread keys permit a wider base, prevent any tendency of the key to tip. This results in a stronger key, a more simple hub design.

4. Shorter over-all length.

Like all Twin Disc Machine Tool Clutches, the single hub clutch has phosphor bronze driving plates driving high carbon, heat-treated and ground steel driven plates to assure excellent wear, a smooth engagement without clutch chatter. The grooves cut in the driven plates, shear the oil out quickly. This action provides maximum capacity with minimum pressure. Adjustments as close as .003″ are easily and quickly made by hand at a single point, without the use of special tools.



ILLUSTRATED: *Cross-sectional view of Twin Disc single hub MT Clutch.*

TWIN DISC
CLUTCHES AND HYDRAULIC DRIVES
REG. U.S. PAT. OFF.

TWIN DISC CLUTCH COMPANY, 1325 RACINE STREET, RACINE, WISCONSIN

Co., New Haven, Conn., is claimed to be particularly important in view of the shortage of nickel salts for black nickel solutions. The deposits of this process are hard, black and very adherent. A solution of 12 ounces to a gallon of water is used, operated from 140 to 190 degrees Fahr. Current densities required are low, ranging from ½ to 5 amperes per square foot.

Engineering Dept. Equipment

Speedy Whiteprint Machine

ESPECIALLY designed to meet today's heaviest print-making demands, Model "B" Whiteprint machine manufactured by Ozalid Products Division, General Aniline and Film Corp., Johnson City, N. Y., can produce finished whiteprints at speeds up to 20 feet per minute. Such features as synchronized printing and developing, permitting use of continuous yardage as well as cut sheets, a temperature control for printing cylinder, and a special light conveniently placed above print-receiving tray, enable operator to check prints for correct printing and developing speeds as they are made. The machine delivers prints in the front or rear,

and has an adjustable burner shade to permit running prints of varying opacity without changing printing speed. An automatic air pickoff and an effective blower hook-up are other features of the machine. Prints are made by exposure and dry developments, original drawing being placed on light-sensitive material and fed into the machine where it is exposed to a mercury vapor lamp for printing. It then is conveyed automatically over a developing tank and dry-developed when material comes in contact with ammonia fumes.

Safe Speeds for Flywheels

(*Concluded from Page* 44)

except for the parts near the ends of the cylinder.

In a manner similar to the above examples, the allowable speed can be obtained by first determining the element for which the distortion energy is a maximum. The value of the distortion energy for the three-dimensional case of stress is

$$V=c(S_r^2+S_t^2+S_z^2-S_rS_t-S_rS_z-S_tS_z) \quad \ldots\ldots (u)$$

Substituting the values of the stresses in Equation u, the distortion energy is

$$V=cb^4\Big\{ c_1^2\Big[2+\alpha^2(7-3c_3)+2\alpha^4\Big]-2c_1c_2(1+\alpha^2)^2- x^2\Big[c_1^2(1-\alpha^2)(1+c_3)+4(\alpha^2+1)- c_1c_2(5+5\alpha^2+c_3+c_3\alpha^2)\Big]+x^4\Big[c_1^2(1-c_3+c_3^2)+ 2c_2(2c_2-c_1-c_1c_3)\Big]+3c_1^2\frac{\alpha^4}{x^4}\Big\} \quad \ldots\ldots\ldots\ldots (v)$$

where $\alpha = a/b$, $x = r/b$. To determine the value of x for maximum V it is necessary to know the ratio $a/b = \alpha$ and of m. With $\alpha = .5$ and $m = .3$,

$$V=c\left(\frac{\delta}{g}\omega^2b^2\right)^2\left[1.07-4.89x^2+.041x^4+\frac{.034}{x^4}\right] \quad (w)$$

Variation of V with x shows that it is maximum for a value of $x = 1$. The allowable speed is calculated by equating this maximum to the allowable value for simple tension equal to cS_w^2. Making this substitution the allowable speed is

$$\omega=.71\sqrt{\frac{gS_w}{b^2\delta}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

In a similar manner the allowable speed can be obtained for other values of $x = a/b$.

Allowable speeds for rotating disks of variable thickness can also be determined in a manner similar to the above. It is not convenient, however, in these more complicated cases to develop a general theory[4].

[4] For a summary of the stress analysis in disks with variable thickness see, for example, R. J. Roark, *Formulas for Stress and Strain*.

Renewed impetus is being given the development of three-dimensional pictures by a new method utilizing a transparent ridged screen which produces a depth effect similar to that achieved by grandmother's parlor stereoscope in the gay nineties. Viewing glasses essential with the light polarizing method are unnecessary in this new development.

NEW! Second Edition of "Plastics in Engineering"

by John Delmonte

Over six hundred pages of authentic information on plastics. Answers practically all questions that confront designers faced with the necessity of securing information on plastic materials of all kinds.

CONTENTS

Since publication of the First Edition of "Plastics in Engineering", less than two years ago, the plastics industry has undergone rapid advancement . . . resulting in new and important developments for the design engineer working with these materials.

Because national emergency has necessitated withholding, or at least curtailing, use of strategic metals for defense, a minor revolution in the design of machine parts has occurred. The result is that now—more than ever before—manufacturers must become acquainted with ways of adapting plastics to their design program.

A careful study of the contents of "Plastics in Engineering" will show its extremely broad and inclusive consideration of plastics . . . will give an idea as to why it has become the accepted authority on plastics.

New chapters have been added and technical data have been reorganized to permit more easy reference to the text material. Extruded plastics and resin-bonded plywoods, for example, are given more prominence in the Second Edition because of tremendous expansion in these developments during the past year.

For up-to-date information on the subject of plastics as applied to engineering and the design of machines and machine parts, order a copy of this recognized authority today.

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what higher than those obtained for a single spring.

For a three-spring nest a similar analysis shows that the energy stored is given by

$$U = C_V'' \frac{S^2 V}{4G} \quad \text{(22)}$$

where

$$C_V'' = C_V \left[1 + \left(\frac{c-1}{c+1} \right)^2 + \left(\frac{c-1}{c+1} \right)^4 \right] \quad \text{(23)}$$

Values of C_V'' plotted against c in *Fig.* 3 show that for a three-spring nest under the assumptions of equal deflection and equal solid height, the maximum energy is stored in a given volume of space for springs having indexes between 6 and 9.

For a spring nest subject to *static* loading, the analysis may be made in exactly the same way as before, except that the stress S_S is figured from Equation 10, which neglects the stress augment due to curvature, and instead of C_V the factor C_S in Equation 12 is used. This analysis gives:

1. For a two-spring nest, statically loaded

$$U = C_S' \frac{S_S^2 V}{4G} \quad \text{(24)}$$

where

$$C_S' = C_S \left[1 + \left(\frac{c-1}{c+1} \right)^2 \right]$$

2. For a three-spring nest, statically loaded

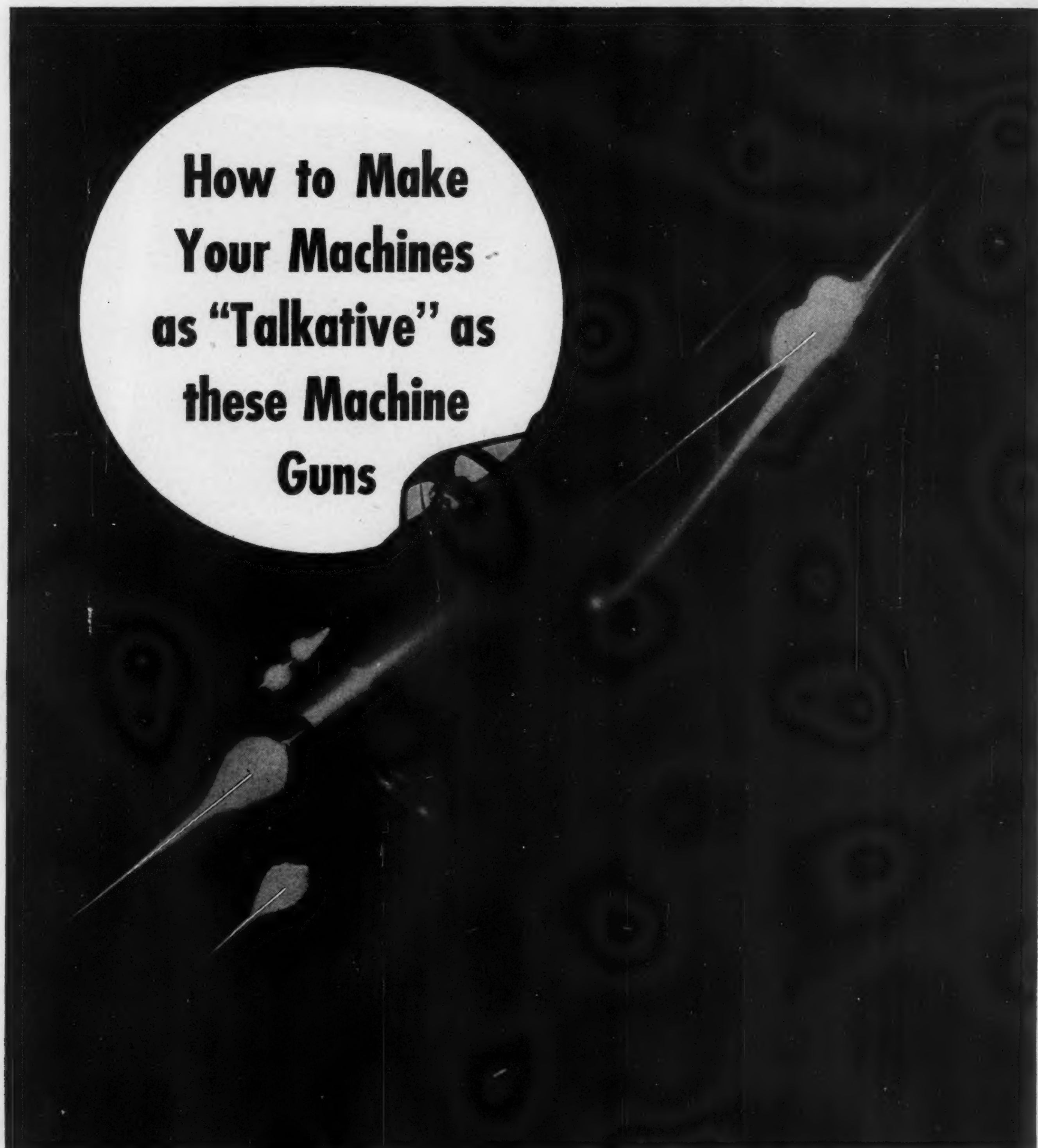
$$U = C_S'' \frac{S_S^2 V}{4G} \quad \text{(25)}$$

where

$$C_S'' = C_S \left[1 + \left(\frac{c-1}{c+1} \right)^2 + \left(\frac{c-1}{c+1} \right)^4 \right] \quad \text{(26)}$$

Values of C_S' and C_S'' are plotted against spring index c in *Fig.* 4. From these it appears that for *static* loads the maximum energy storage in a given space will be obtained by using springs with indexes around 3 to 4 for a two-spring nest and with indexes of 4 to 5 for three-spring nests. These optimum values of spring index are considerably lower than the corresponding ones for variable loading. They are also higher than those for a single spring statically loaded.

It should be noted that, in practical design, a great many factors other than those mentioned may require consideration. However, where the problem is principally one of obtaining maximum energy storage (or maximum deflection for a given load) within a given space, the curves of *Figs.* 3 and 4 and the conclusions based thereon should prove useful as a guide to the design.


How to Make
Your Machines
as "Talkative" as
these Machine
Guns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YOU bet these "3C" Bulletin 102 Heavy Duty Snap Action Track Type and Pilot Circuit Crane Hook Limit Switches are packed full of service.
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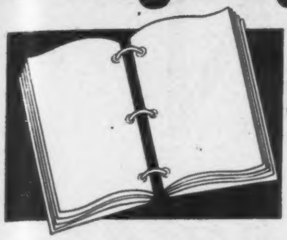
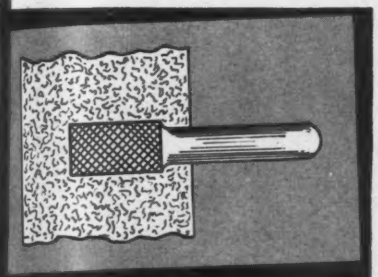
Designing Molded Plastics Parts: *INSERTS*



from the engineering files of One Plastics Avenue

To incorporate inserts in molded plastics parts the following general factors should be considered: (1) Avoid use of delicate inserts for compression type molds. (2) Use simple shaped inserts, preferably round, for easy cleaning. (3) Minimum amount of plastics around insert: 3/32 in. for ¼ in. insert; ⅛ in. for ½ in. insert; ¼ in. for inserts over ½ in. (4) Keep inserts from corners and edges. (5) Placement of inserts should be at right angles to parting lines. (6) Knurl, punch or groove inserts for good anchorage. (7) Where metal inserts are used as electrical contacts, provide proper air gaps at point of arcing to avoid burning or carbonization of plastics. (8) If possible, avoid using inserts for injection molded thermoplastics.

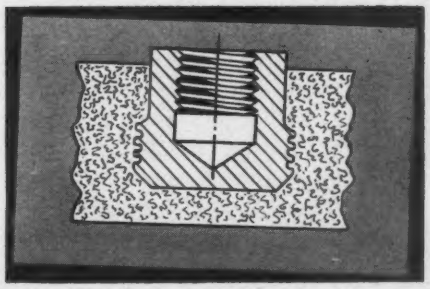


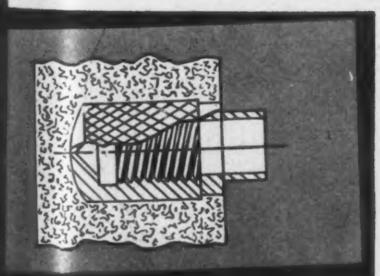
◀ ROD AND WIRE INSERTS

The best general length to diameter ratio is 2 to 1 wherever practicable. For compression molded parts the maximum ratio of length to diameter of rigid inserts should be 4 to 1 if one end is supported, 8 to 1 if both ends are supported.

THREADED INSERTS ▶

The preferred ratio of length to diameter for all threaded inserts is 2 to 1. Always use a coarse knurl for rag compounds, and as coarse as possible for other compounds. Where heavy loads are encountered design insert to carry the stress; do not put load on plastics part. Female inserts are generally retapped after molding. Length of thread should be kept to minimum and projecting end of insert should be smooth, round and free from knurl.



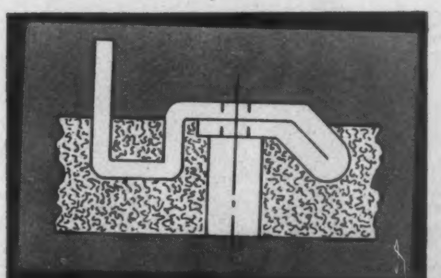


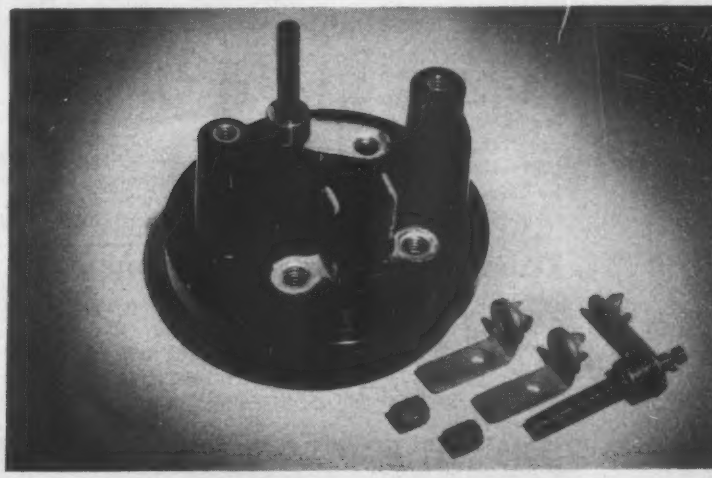
◀ SPUN-OVER INSERTS

For anchorage, dimensions, materials, same rules apply as to threaded inserts. This is a good method of assembling contact strips, washers, etc., on the molded part.

STAMPED OR PUNCHED ▶ INSERTS

Avoid delicate or unsupported projections in compound. For rag or fabric compound use rigid inserts whenever possible. Good anchorage is important. Laminated metal inserts may be molded.





TYPICAL APPLICATION Typical of molded parts incorporating inserts is this electrical switch section which has five separate inserts including each of the major types—rod, threaded and stamped.

ADDITIONAL FACTS General Electric has 14 standard female inserts. Inserts of Mycalex, porcelain, woven wire mesh, laminated phenolic material, woven glass, etc., can be molded in plastics parts. Because of current shortages metal inserts should be used only when essential to part. G.E.'s consulting engineering service will advise you on best methods of design.

ONE PLASTICS AVENUE at Pittsfield, Massachusetts, is the headquarters for five plants of the Plastics Department of General Electric Company. It signifies the location of complete plastics facilities for development, material manufacture, designing, engineering, moldmaking, molding and laminating.

REPRINTS of this advertisement may be obtained by writing Section D-1, General Electric Plastics Department, Pittsfield, Mass.

PLASTICS DEPARTMENT

GENERAL ELECTRIC

MEN *Of Machines*

ADDITIONAL duties—those of assistant general manager — have been added to the present ones of Hubert D. Tanner who now is vice president and manager of the machinery department of Pratt & Whitney Division Niles-Bement-Pond Co., West Hartford, Conn. Mr. Tanner was born in 1889 in Pawtucket, R. I., and was educated in mechanical engineering at Brown University. He also attended Rhode Island School of Design. For eight years he was employed as an engineer in both machine and tool designing at Brown & Sharpe. In 1920 he joined Pratt & Whitney as an engineer, and has been with the company continually ever since. He was made manager of the machinery department in 1935, and four years later a vice president. Mr. Tanner is a member of the A. S. M. E., S. A. E. and A. G. M. A.

CONSULTING engineer and inventor of some prominence, holding many U. S. patents, R. E. Bressler has been made vice president and general manager of Kol-Master Corp., Oregon, Ill. About 1929 Mr. Bressler was called by a large Chicago manufacturer to check a stoker design and as a result of that study evolved and invented the Kol-Master automatic stoker. The Kol-Master Corp. was then organized to build this

stoker, on which Mr. Bressler holds four patents. These are the hopper which is "offset" so that it does not block or cover fire door; the feed screw, conveying coal from hopper to fire, which has patented reverse flight which "feeds back" coal in fire-bed; burner or retort in which Mr. Bressler's patents embrace "dividing plates"; and an automatic combustion control, which by wind-box pressure automatically controls and synchronizes the feed of both fuel and air. Previous to his appointment as vice president and general manager, Mr. Bressler was chief engineer.

♦

E. C. SPARLING has been appointed chief engineer of Sperry Gyroscope Co., Brooklyn.

♦

CAMERON N. LUSTY, assistant chief engineer, Taylorcraft Aviation Corp., Alliance, O., has been appointed chief engineer of Mercury Aircraft Co., Menominee, Mich. He succeeds J. B. BAUMANN who resigned to join the engineering staff of Frankfort Sailplane Co., Joliet, Ill.

♦

HAROLD M. LOCHRANE, designing and operating engineer, recently joined Kaydon Engineering Corp., Muskegon, Mich., in charge of methods and standards for the company.

♦

E. M. MAY has been elected president, Midwest Stoker association, to succeed MOUNT BURNS, resigned. Mr. May is manager, Combustioneer division, Steel Products Engineering Co., Chicago.

♦

A. P. FONTAINE succeeds E. G. BRUCE as chief engineer of the engineering and development department of Vultee Aircraft Inc. Mr. Bruce became the company's chief research engineer. Mr. Fontaine was formerly chief engineer of the company's Wayne, Mich., plant.

♦

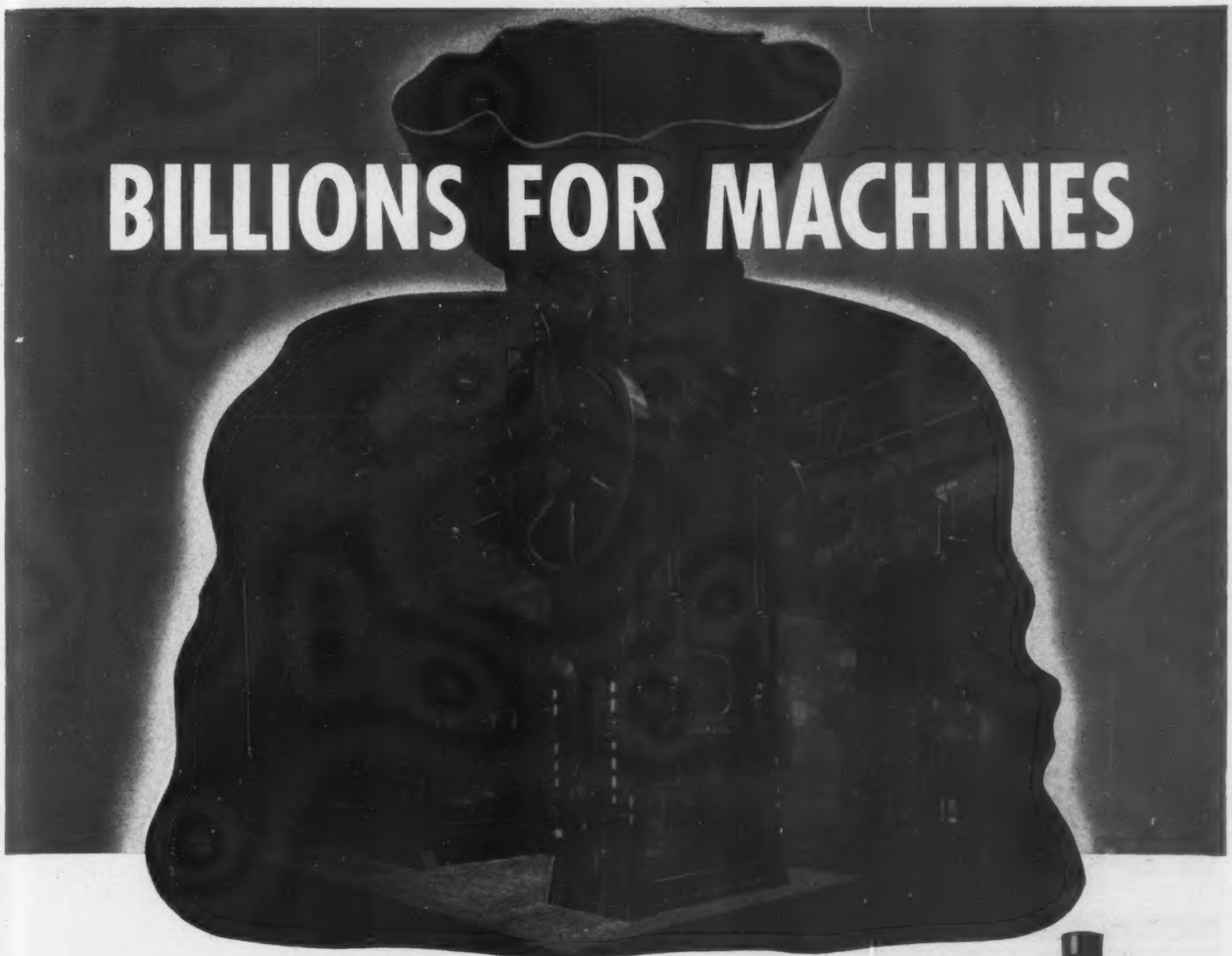
COL. HERBERT W. ALDEN was recently awarded the Col. Frank A. Scott gold medal for meritorious service to industrial preparedness by the Army Ordnance association. He served as chairman of the ordnance automotive advisory committee, and is director of engineering of Timken-Detroit Axle Co.

♦

N. E. WAHLBERG, vice president, Nash-Kelvinator Corp., will head a newly-formed engineering research division of the company. MEADE F. MOORE, formerly chief engineer of the Nash Motors division, will become chief research engineer of the new division; FLOYD KISHLINE, of the engineering department, will become chief engineer.

♦

HARRY A. WINNE, formerly assistant to the vice president in charge of general engineering operations, has been made one of the five new vice presidents recently appointed by General Electric Co.,



Produce no Defense Standing Still

—and they *will* stand still if some bearing, somewhere, sometime, gets no lubricant.

Make the small additional investment to insure that all bearings, in all places, in all the machines you build, will be lubricated. Install Farval!

The Farval System will deliver lubricant in exact measured quantities to a group of bearings from one central station, as frequently as desired. Every bearing receives the amount of lubricant it requires and — not a bearing is missed.

Farval will protect your Customers' present production schedules—quickly pay back the small additional investment—and continue to earn for them in the future.

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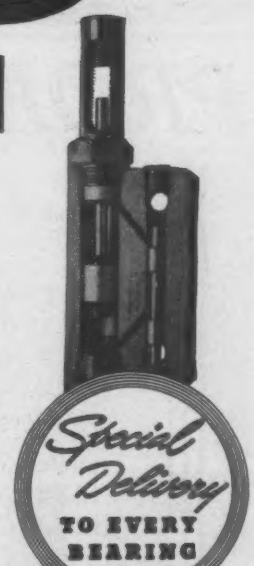

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Schenectady, N. Y. Mr. Winne will be in charge of design engineering, apparatus department.

♦

DR. ROBERT YARNALL was honored by receipt of the Hoover medal at the recent annual meeting of the American Society of Mechanical Engineers.

♦

JUAN J. TRIPPE, president of Pan-American Airways, has been awarded the 1941 Daniel Guggenheim medal for "the development and successful operation of oceanic air transport."

♦

I. J. KAAR has been named managing engineer of the receiver division of General Electric's radio and television department, Bridgeport, Conn. Since November, 1934, he has been designing engineer of the division.

♦

ALBERT WARTINGER has been named chief engineer of Sheffield Corp., Dayton, O. He was formerly chief engineer of the tool and die division of the company, with which he has been connected for 15 years. FAY ALLER, formerly chief engineer of the gage and machine tool division, has been named director of research.

♦

ROBERT M. KALB, for 13 years research engineer of Bell Telephone Laboratories Inc., New York, has been appointed assistant chief engineer, Kellogg Switchboard & Supply Co., Chicago.

♦

HAROLD SINES VANCE has been elected to the board of trustees of the Illinois Institute of Technology. He is chairman of the board of directors of Studebaker Corp.

♦

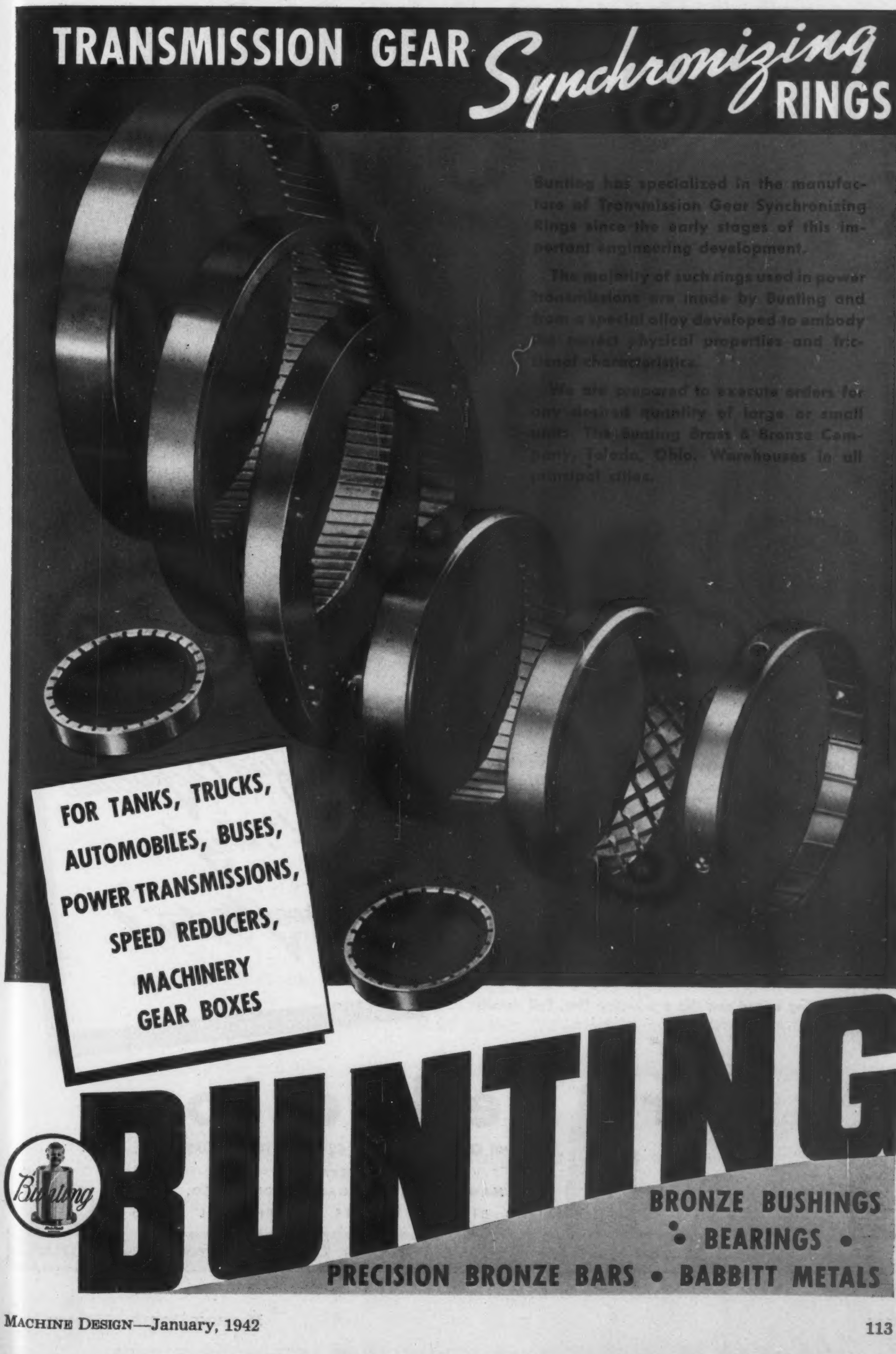
A. W. S. HERRINGTON, president of Marmon-Herrington Co., Inc., Indianapolis, has been chosen as national president of the Society of Automotive Engineers for 1942.

♦

O. K. MARTI has been made consulting engineer of Allis-Chalmers Mfg. Co. rectifier division. H. WINOGRAD succeeds Mr. Marti as engineer in charge of rectifier design.

♦

DR. ARTHUR N. TALBOT was recently honored by the American Railway Engineering association, when a bronze tablet bearing his name and recognizing his nearly thirty years of research work for railroads, was presented to the University of Illinois for erection in the laboratory bearing his name. Dr. Talbot is professor emeritus, University of Illinois, Champaign, Ill. The presentation of the tablet marked his retirement as chairman of the association's special committee on stresses, of which he became head in 1914.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Saved Time

Conserved Material

Bronze inserts were formerly installed in marine valves of this type to insure long trouble-free service. Now a faster, simpler, less costly method does the job equally as well, if not better. Utilizing the Airco oxyacetylene flame and bronze welding rods, a "sure sealing" bronze seat is deposited on the valve. Welding eliminates a step in manufacture, saves time, conserves a valuable defense metal.

Simplified product design is often achieved by welding. With this in mind Air Reduction makes available its broad experience to assist its customers wherever possible, both on the drafting board and the production line. Full details on request.

Air Reduction

General Offices: **60 EAST 42nd ST., NEW YORK, N. Y.**

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MAGNOLIA-AIRCO GAS PRODUCTS CO.

AIRCO DISTRICT OFFICES IN PRINCIPAL CITIES

AIRCO — WILSON

Anything and Everything for **GAS WELDING or CUTTING and ARC WELDING**

WESTINGHOUSE
ELECTRIC & MANUFACTURING COMPANY

→ W.G.A.
Bill - check up on this as soon as you can R.D.

EAST PITTSBURGH, PA.

January, 1942

Only One in a Hundred Men
Will Be Interested in This Letter.
Are You That One?

Gentlemen:

This letter concerns a new material - "Prestite" - a compression-molded vitrified ceramic which possesses the outstanding advantages of industrial glasses and molded plastics. Our purpose is to suggest to those responsible for design, the breadth and scope of "Prestite's" potential applications and its possibilities as a replacement for difficult-to-obtain materials. Here are the facts.

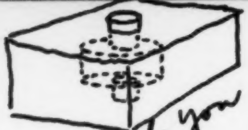
"Prestite" has appreciably greater resistance to breakage than either porcelain or industrial glass. In compressive strength, it rivals alloy steels. 600° C. doesn't faze its mechanical strength. It is completely resistant to all corrosives except hydrofluoric acid and is impervious to moisture. It may be glazed in strikingly beautiful, ceramic colors. It exceeds ordinary porcelains and plastics in dielectric strength.

Priorities Solution

What do you think?

But the fact that makes "Prestite" pregnant with opportunity is the chance it gives designers to provide, in a material that can be molded as freely as organic plastics, the above properties ... in intricate shapes, to tight tolerances, on a mass production basis.

An inorganic plastic!



Did you catch this, Bill?

Furthermore -- "Prestite," like cast metals, can be cored to produce intricate interior cavities in the finished piece ... and a way has been found to intimately join "Prestite" to metals, "surface-to-surface."

"Prestite's" first use was as a dielectric ... and innumerable electrical insulators have been made from it. More recently its use has spread far beyond this field -- to such unusual applications as valve seats for high-speed pumps, sandblast nozzles, coffee grinder plates, dental machine instrument panels, and water cooler storage compartments.

Must be pretty tough!

Such applications amply illustrate the extraordinary uses of "Prestite" ... suggest that it may be the answer to your critical shortage ... or the one material needed to create new products for new markets.

If this brief discussion of "Prestite" suggests its use in one of your designs, write for our new booklet covering "Prestite" applications -- or, if you wish, our skilled application engineers will be glad to serve you.

Sincerely,

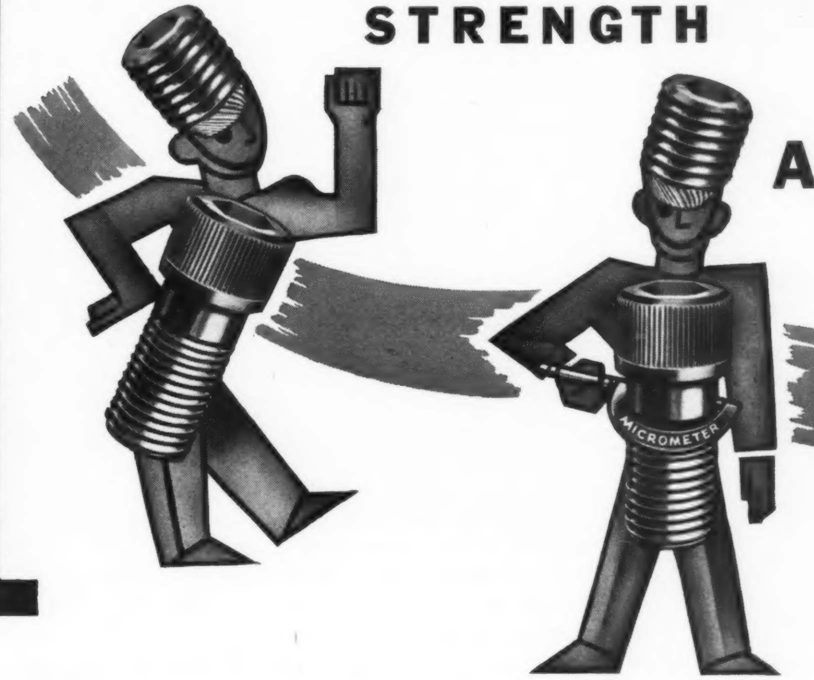
Westinghouse

Westinghouse Electric & Manufacturing Co.

Westinghouse Electric & Manufacturing Co.
Department 7-N
East Pittsburgh, Pa.

Drop them a note, Bill.
Let's get some more dope

STRENGTH



ACCURACY

plus KNURLING

"UNBRAKO" KNURLED SOCKET HEAD CAP SCREWS

Strength? . . . it's the "Unbrako" middle name, and there's lots to spare.

Accuracy? . . . you bet! Micrometric uniformity is characteristic.

And the KNURLING? . . . ah, there you have it! The big "Unbrako" feature that gears smooth fingers to oily heads and saves time by speeding up assembly.

Convinced? . . . a trial order will prove it to you. Place it now for KNURLED "UNBRAKO".

"UNBRAKO" SELF-LOCKING SET SCREWS

Just what you want to end ordinary set screw sorrows! With "Unbrako" Self-Lockers, just tighten them up as usual and the KNURLED points dig in for the duration and stay locked . . . no matter how severe the vibration and number of applications.

Try a batch of these too!

The KNURLING of Socket Screws originated with "Unbrako" years ago.

UNBRAKO
Reg. U. S. Pat. Off.

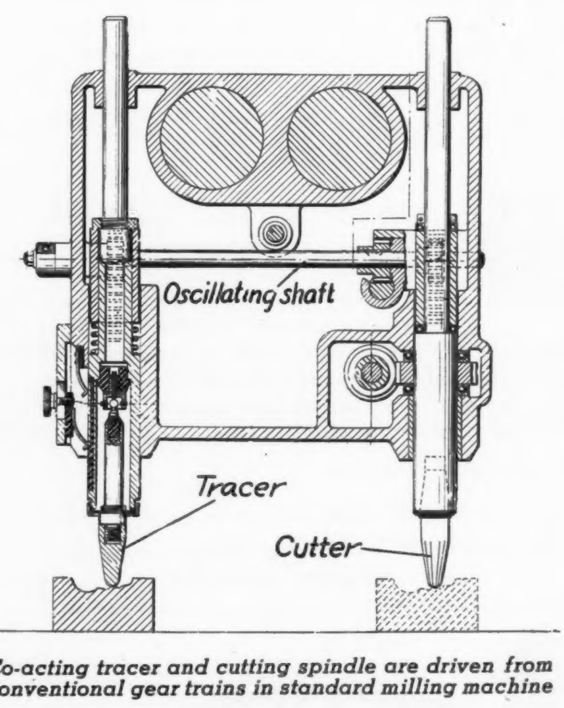
STANDARD PRESSED STEEL CO.
JENKINTOWN, PENNA. BOX 102
— BRANCHES —
BOSTON • DETROIT • INDIANAPOLIS • CHICAGO • ST. LOUIS • SAN FRANCISCO

Noteworthy PATENTS

Combines Control Systems

MACHINE tools for copying operations in which irregular or involved surfaces are machined ordinarily are built to perform only this type of function. A patent for an attachment to a conventional milling machine which will enable it to carry out such copying operations has been assigned to the Kearney & Trecker Corp.

Complete details of the design are beyond the scope of this explanation. However, the manner in which electric, hydraulic and mechanical controls have been utilized in the design will be of undoubted interest. Shown in the first illustration are the essentials of the operating parts of the machine attachment. The tracer spindle follows the contour of a pattern as the table of the milling machine traverses both the pattern and the work blank. As will be discussed in some detail in connection with the second illustration, when the tracer spindle encounters a change in contour of the pattern the

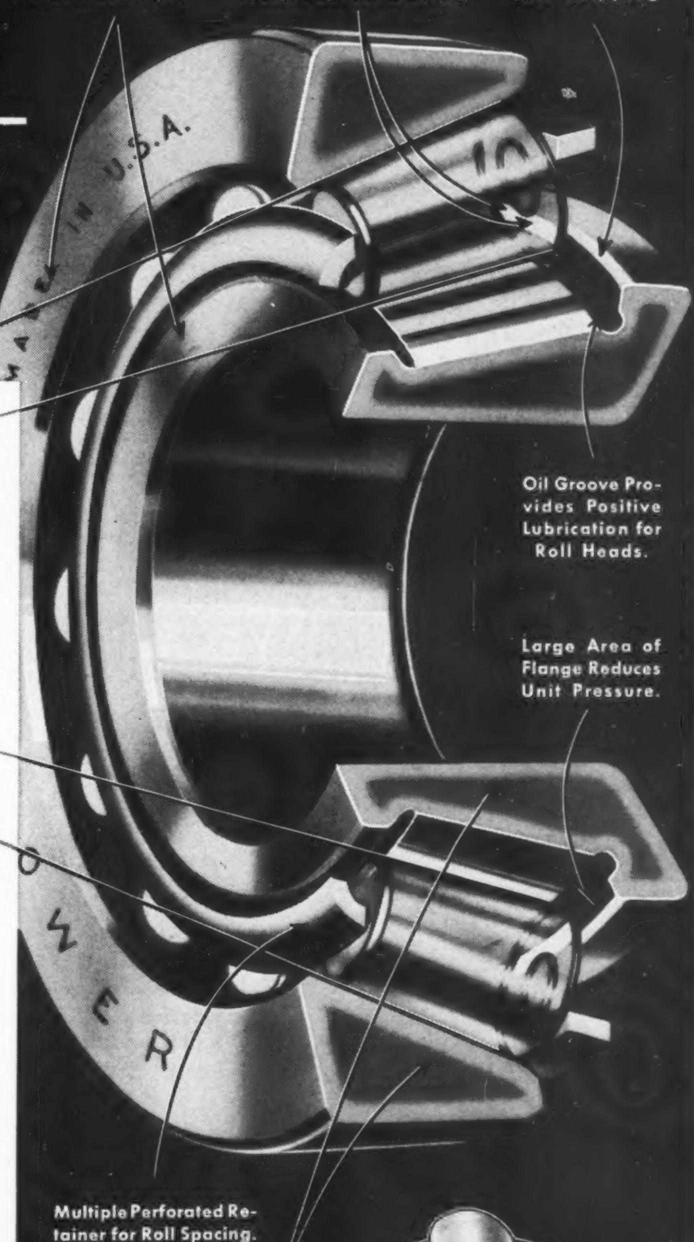


Co-acting tracer and cutting spindle are driven from conventional gear trains in standard milling machine

oscillating shaft effects an immediate raising and lowering of both the cutter and the tracer. The essence of the control system lies in the manner in which time delay in the response to contour changes has been reduced to an absolute minimum.

The oscillating shaft is driven through a pinion

OF FIRST IMPORTANCE—*Design*



THIRTY-EIGHT years of Roller Bearing manufacture have proved that design is the most important element in a bearing.

Likewise, the use of many millions of Bower Roller Bearings as original equipment over a period of many years in America's leading large-production automobiles has proved the correctness of BOWER DESIGN.

Bearing users will appreciate that the exacting standards of the automotive industry and the severe usage of bearings in automobiles offer a challenge that no roller bearing can meet unless it possesses the highest degree of quality known to the bearing industry.

One of the secrets of Bower's leadership is the fact that its technical men have never waited upon the ingenuity of other men. Bower engineers push relentlessly ahead—far beyond the needs of the moment—to make new technical discoveries and to apply them always in ADVANCE.

This Tapered Roller Bearing is a leading example of Bower design. It embodies important advantages that no other bearing possesses—advantages that Bower engineers discovered and incorporated ahead of all others.

For more detailed information on Bower design, ask us for a copy of the folder, "Secrets of Bower Roller Bearing Design and Quality."

Bower Finish Like A "Face-Lifting" Operation

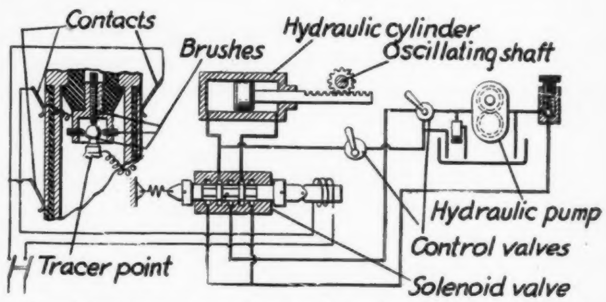
1. ROUGH GROUND—Photomicrograph—25 diameters—showing amorphous film with roughness of approximately 15 micro-inches (millionths of an inch). **2.** FINISH GROUND—More but finer scratches—surface finish of approximately 10 micro-inches composed of amorphous metal left by finish grinding—Photomicrograph of 25 diameters. **3.** FINAL FINISH—25-diameter photomicrograph showing amorphous metal and grinding marks removed, baring hard surface and smoothness of approximately 3 micro-inches—scratches below surface.

BOWER ROLLER BEARING CO.
Detroit, Michigan

engaged with a rack on the piston rod of the hydraulic cylinder shown in the second illustration. Hydraulic fluid is supplied to this cylinder from a pump through two control valves and a five-way, spring-opposed, solenoid-operated controller.

Formed on the upper end of the tracer point is a spherical electric contact. This sphere is surrounded by five adjustable co-operating contacts, four in the horizontal plane and one located directly above. These contacts are electrically connected to two commutator strips mounted on the tracer head and reciprocally movable with it. Stationary brushes in contact with these strips effect the operation of the solenoid valve.

When a contour change is encountered by the tracer point, contact is established in the solenoid circuit which causes the solenoid valve piston to



Combination of mechanical, hydraulic and electric controls provides utmost sensitivity in profiling attachment for milling machine

be moved to the right against its opposing spring. Pressure is thereby introduced to the left end of the hydraulic cylinder, moving the piston to the right and raising the cutter and tracer spindles. After a preselected movement of these spindles, contact of the brushes with the commutator strip is broken and the solenoid circuit de-energized. The spring then immediately returns the valve piston to the left, introducing fluid to the right end of the hydraulic cylinder and thus moving the spindle downward on a cutting stroke. The control valves are so adjusted that the upward stroke of the spindle occurs at high speed while the downward or cutting stroke is set to the permissible cutting speed.

If contour changes are abrupt the lower, left-hand brush may be adjusted with respect to the upper one in such a manner that the downward or cutting stroke of the spindle will be automatically limited to less than that necessary to generate the finished contour. In this way excessive load on the cutting tool may be avoided.

Measures Acceleration

INTRODUCTION of errors in reading, resulting from nonuniform scale graduations, is obviated by a mechanical compensation arrangement in the accelerometer illustrated. Also the use of a pair of acceleration-responsive masses which are interlocked mechanically eliminates errors which might be introduced by rotation about any longitudinal horizontal axis. The patent for the instrument is

"Watchmen"

that Protect Your Defense Assemblies!

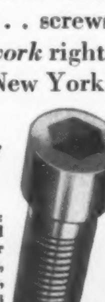
THESE QUALITY-CONTROL RECORDS PREVENT "DOUBTFUL" SOCKET SCREWS THAT CAUSE DELAYS AND REJECTS

From careful analysis of the raw material, through every step in production, *each lot* of Parker-Kalon Socket Screws is guarded by these "watchmen" . . . the quality-control records you see here. One reports on the Parker-Kalon Laboratory's analysis of the special alloy steel . . . another is the Lab's point-by-point okay of all physical characteristics of the finished screws. A third record covers an extensive series of critical inspections of dimensions and other details.

The uniformly high standard of quality which MUST RESULT from such rigid control is ample reason why industry specifies PARKER-KALON for important defense assemblies. Besides, it costs no more to have this protection against "doubtful screws" . . . screws that *look* all right but some of which fail to *work* right. Parker-Kalon Corp., 192-200 Varick Street, New York.

Quality-Controlled

16-point test and inspection routine covers: Chemical Analysis; Tensile and Torsional Strength; Ductility; Shock Resistance under Tension and Shear; Hardness; Head diameter, height and concentricity; Socket shape, size, depth and centricality; Class 3 Fit Threads; Clean-starting Threads.



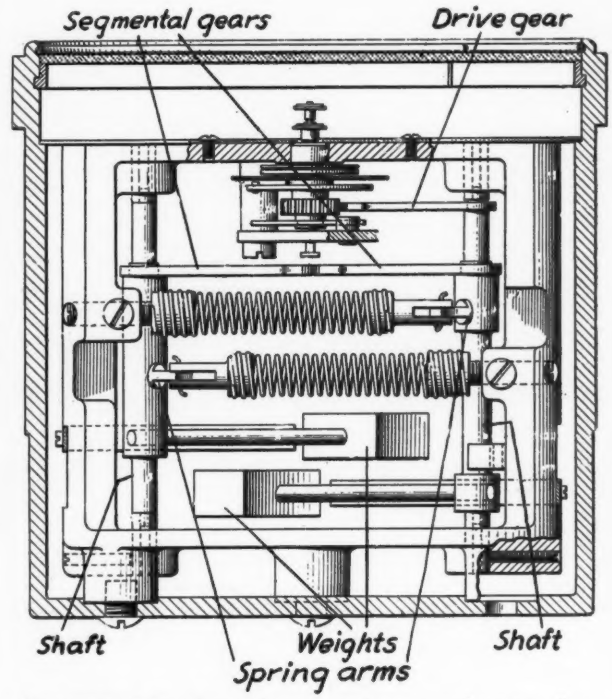
PARKER-KALON
Quality-Controlled
SOCKET SCREWS

Give the Green Light to Defense Assemblies

assigned to the Square D company.

Two essentially identical acceleration-sensitive mechanisms are used, each mounted on one of a pair of vertical shafts. A pair of weights or masses is fastened to the ends of arms which extend radially from the shafts. Also extending radially from each shaft is a spring arm whose angular displacement from the weight arm is predetermined. Each spring arm is fastened by means of a clevis to the end of a tension spring, the other end of which is anchored to the instrument housing. Also integral with the vertical shaft are intermeshing segmental gears which constrain the shafts to oscillate only in unison.

It will be evident that if the instrument is accelerated positively in a straight line perpendicular

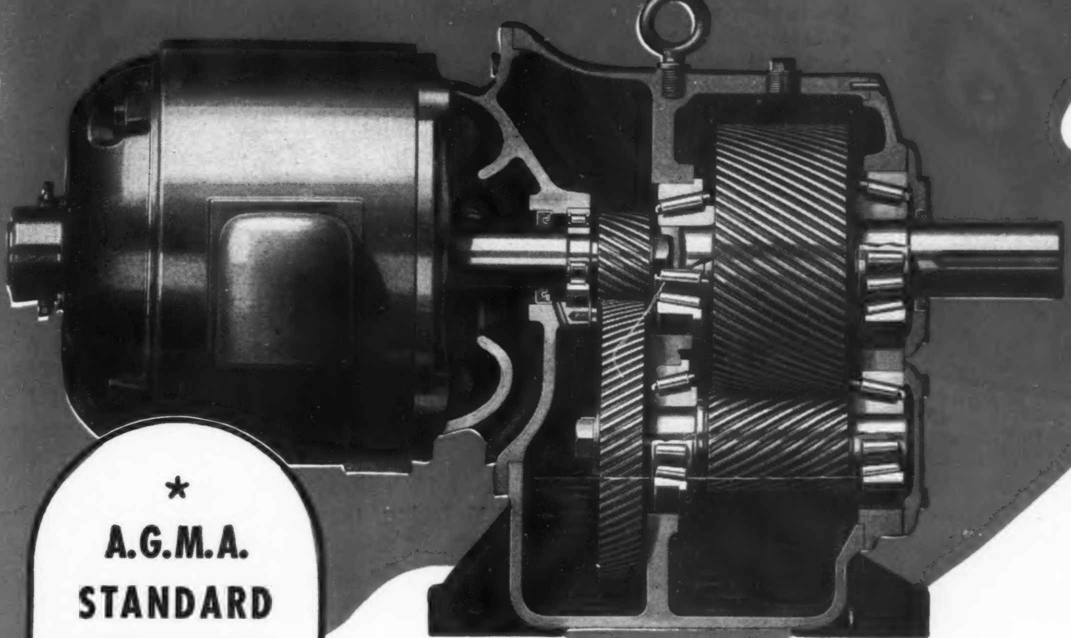


Compensation for tilting and a mechanism for obtaining equal scale graduations are provided mechanically in accelerometer

to the illustrated section, the weights will tend—by reason of their inertia—to resist the change in velocity. They will, therefore, rotate the vertical shafts with respect to the housing. However, as the mass is moved in a direction opposite to that of the imposed acceleration their effective lever arms, about the vertical shafts, change. If uncompensated this effect would result in unequal calibration of the dial. However, the spring arms are so located that as the shaft rotates under the impetus of the weight the spring torque arms change in proportion to the change of the weight arm, thus providing automatic compensation.

If the instrument is tilted about an axis perpendicular to the illustration no effect on the instrument reading will be noted. This is accomplished by the fact that when such tilting occurs each weight tends to rotate its own shaft in the same direction as the other. However, the two shafts are co-ordinated by means of the intermeshing seg-

COMPACT... EFFICIENT... DURABLE...



★

A.G.M.A.
STANDARD
OUTPUT SPEEDS

STANDARD OUTPUT SPEEDS for concentric and parallel shaft integral hp motorized reducers. Based on 1750, 1430 or 1165 rpm motor operating speeds.

1430	190	25
1170	155	20
950	125	16.5
780	100	13.5
640	84	11.0
520	68	9.0
420	56	7.5
350	45	6.0
280	37	5.0
230	30	4.0

AGMA

THE LINK-BELT MOTORIZED REDUCER

● If your idea of a perfect drive unit is one you can install with complete confidence—relying on it for day-in day-out service—then you're thinking of the Link-Belt motorized helical gear speed reducer. Motor and gear set are as near a unit as it is possible to achieve, and quiet operation and permanent alignment are assured without coupling or base plate. Heavy-duty bearings on low speed shaft carry large overhung loads. Motor and high speed gears completely accessible without disturbing low speed gears or driven machine. Available in 8 standard sizes, double and triple reduction types—up to 75 H. P., for mounting on wall, ceiling or floor. Book No. 1515 gives further details of this rugged, compact unit—send for it.

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Leading Manufacturer of Mechanical Transmission Equipment—Silent and Roller Chains . . Speed Reducers . . Speed Variators . . Roller, Ball and Babbitted Bearings . . Collars . . Couplings . . Base Plates . . Take-Ups . . Clutches . . Gears . . Sprockets . . Hangers . . Shafting . . Pulleys, etc.

LINK-BELT SPEED REDUCERS

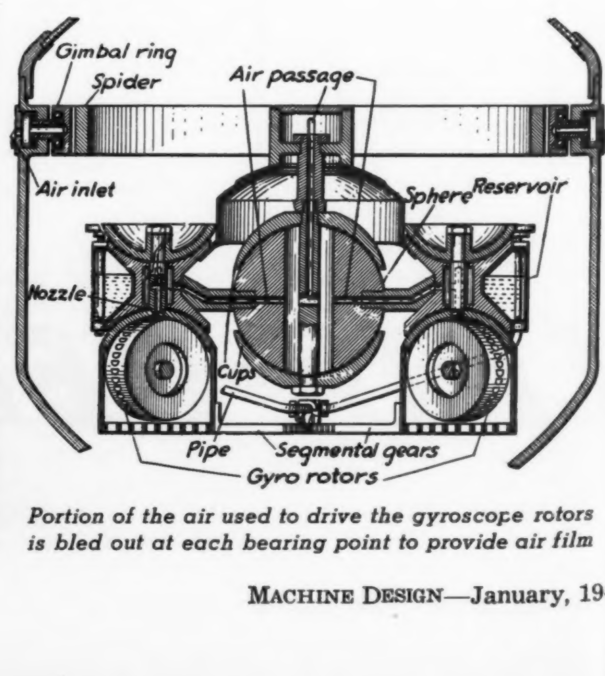
mental gears and hence can rotate only in opposite directions. In this way compensation for tilting is achieved. The right-hand vertical shaft has affixed to its upper end a second segmental gear engaging a pinion which drives the indicating pointer over a dial.

Compass Floats on Air

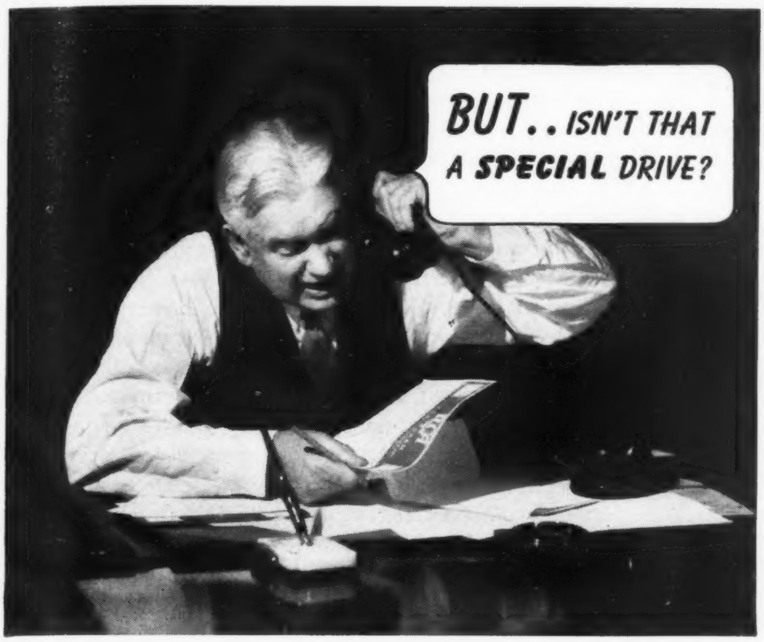
MANNER in which air is utilized to provide substantially frictionless bearing support as well as the driving medium for the rotor is of interest in the gyro compass the patent for which is assigned to the Sperry Gyroscope Company. The problem of supplying air to the working parts of such a mechanism is complicated by the fact that the sensitive element of the gyro compass must, of course, be provided with two degrees of freedom about mutually perpendicular axis in a horizontal plane.

The gimbal ring houses two diametrically opposed ball bearings, each of which bears on a hollow pin integral with the housing of the compass. Thus the first axis of freedom is horizontal and in the plane of the paper. Spaced 90 degrees from each of these gimbal bearings is another set of bearings and hollow pins which support the central spider ring, thereby providing a second horizontal axis of freedom perpendicular to the page. At its center the spider supports the entire working element of the gyroscope compass.

Air is introduced through orifices in the outer casing through the hollow pins in the bearing of the gimbal ring, then is led through ducts in the gimbal ring to the bearing assembly of the spider ring. It then flows through two radial spokes to the center of the spider where it enters a housing which supports the main gyro assembly. Fastened to the center or hub of the spider is a spindle containing a concentric drilled hole which acts as an air passage. Rigidly attached to this spindle are two opposed hemispherical cups which support between them a spherical universal bearing. Ample



Portion of the air used to drive the gyroscope rotors is bled out at each bearing point to provide air film



He Was Worried-

*THE REX MAN

–UNTIL I* TOLD HIM HOW WE MAKE "SPECIAL" DRIVES STANDARD!

① **"WHY NOT USE** a Rex Chabelco chain belt for that new heavy drive of yours?" "Here's a *standard* chain belt that has what your 'special' drive demands—high strengths and the ability to absorb great and sudden shocks."

② **CHABELCO HAS MANY** advantages over ordinary chain belts: it's strong, built for high-speed service and strength! Though your drive is new, there are many types of Chabelco chain belts which will do the job.

③ **WE CAN HELP** you speed up your drive design work! These Rex Chabelco steel chain belts are standard. You can get them in a hurry with your priority! You know how important that is these days. For chain belt information—

④ **LOOK AT CATALOG NO. 444!** It describes over 2,000 types and sizes of chain belts and includes 16 pages of copyrighted shaft center distance tables! You'll find its 768 pages of chain belt design and application data to be invaluable!

⑤ **PUT THESE MEN TO WORK FOR YOU!** This Rex engineering department is actually an addition to *your own Engineering Department.* And in days like these, you can use Rex engineers to good advantage in speeding up your work.

⑥ **"YES, WE'VE SPEEDED UP PRODUCTION** and helped our design, too—because of the assistance we received from Chain Belt Company in applying a standard Chabelco chain belt to a drive that was formerly 'special.'"

Rex chain belts are hard at work these days on machines and in plants the country over. They've answered a variety of problems for others; they can do the same for you. Naturally, orders bearing highest priority distinction take precedence. Chain Belt Company, 1643 W. Bruce Street, Milwaukee, Wisconsin.

REX CHAIN BELTS

CHAIN BELT COMPANY OF MILWAUKEE

Baldwin-Duckworth Chain Belt Division, Springfield, Massachusetts • Worcester, Massachusetts
Rex Chain Belt and Conveyor Divisions, Milwaukee, Wisconsin

REX CHAIN BELT COMPANY

clearance is provided between the spindle and the wall of a bore in the sphere so that the rotor assembly may tilt within limits about any horizontal axis without interfering with the spindle. Opposed arms each fastened to the sphere on one end and to the gyro-supporting spools on the other afford air passages for supplying the driving medium to the rotors and also to the hemispherical bearings for the vertical rotor axes. The gyro rotors are mounted on horizontal axes journaled in bearings in individual rotor housings. The bases of these housings support meshing segmental gears which permit mutual angular displacement of the rotor axes. Opposite fluid reservoirs inter-connected by a pipe provide dampening of any pendulum action of the assemly, resulting in a period for the unit which may be as long as 85 minutes.

Air in flowing through the various air passages arrives finally at the nozzle and drives the gyroscopes by means of turbine cups on the rotors. In addition, some of the air is bled out at each of the six hemispherical bearing surfaces and in flowing between these co-operating surfaces provides an air support for all of the critical axes.

Designing Springs for Space Limitations

(*Continued from Page* 59)

the inner spring is equal to the inner diameter of the outer spring, i.e., $2r_1 - d_1 = 2r_2 + d_2$, the following expressions are obtained by using Equation 6:

$$V_1 = \frac{\pi}{4}(n_1d_1)(2r_1+d_1)^2 = \frac{\pi}{4}nd_1^3(c_1+1)^2$$

$$V_2 = \frac{\pi}{4}(n_2d_2)(2r_1-d_1)^2 = \frac{\pi}{4}n_2d_2d_1^2(c_2-1)^2$$

Since $c_1 = c_2 = c$ and $n_1d_1 = n_2d_2$, this gives

$$\frac{V_2}{V_1} = \left(\frac{c-1}{c+1}\right)^2 \qquad (19)$$

Substituting this in Equation 18, the stored energy for a two-spring nest becomes

$$U = C_V' \frac{S^2V}{4G} \qquad (20)$$

where

$$C_V' = C_V\left[\left(1+\frac{c-1}{c+1}\right)^2\right] \qquad (21)$$

and V = volume enclosed by outer spring.

Values of the energy coefficient C_V' are plotted against spring index in *Fig.* 3. From this it is seen that for a two-spring nest under *variable* loading, the maximum energy storage is obtained for spring indexes between 5 and 7. These values are some-

Designing Hydraulic Servo-Circuits

(*Concluded from Page* 56)

servo-motor is inherently strongly damped if the fits at the valve lands and over the working piston are tight, so that leakage is negligible over both (except of course over the valve when it is on dead center). Quite high overshoot velocities will damp out quickly, especially with a four-way valve having lands at least equal in lengths (as in *Figs.* 3 and 4) or longer than, the valve port width. Much longer valve lands than the port width are not recommended since they cause proportionately large stationary errors under load. Valve land shorter than the port width cause faster response, but a higher degree of overshoot. This type of servo will oscillate steadily with a constant amplitude if care is not taken to prevent looseness in the mechanical linkages and appreciable leakage past the working piston.

References

H. L. Hazen—Theory of Servo-Mechanisms, *Journal of the Franklin Institute,* Vol. 218, No. 3, Sept. 1934.

H. L. Hazen—Design and Test of a High Performance Servo-Mechanism, *Journal of the Franklin Institute,* Vol. 218, No. 5, Nov. 1934.

How Metallurgy Affects Deep Drawing

(*Continued from Page* 65)

two numbers hard would be about 40,000 pounds per square inch.

Corrosion resistance of brasses varies with the different compositions. For example, the higher copper or red brasses are more resistant to corrosion by soldering fluxes than the yellow brasses, especially in thin sheets. Intergranular corrosion results from the attack of ammonium chloride which is a constituent of most commercial fluxes. In assembling parts of such thin material, soldering fluxes of a rosin, or a water or alcohol solution of USP zinc chloride are strongly recommended.

In contrast to the corrosion resistance of high copper brasses under ordinary atmospheric conditions, low copper or high zinc brasses are more resistant to corrosion by sulphurous atmospheres. In general if a drawn part is to be subjected to severe corrosive conditions a final anneal after the last drawing operation is desirable. Such an anneal will eliminate incipient intergranular corrosion resulting from the presence of slip lines or other surface defects by restoring a uniform and unstrained crystalline structure.

So-called deep-drawing steels are usually of two types, rimming or killed steel. The former is rolled from an ingot from which the top portion containing the pipe as well as other portions has been cut off by cropping. Killed steel is that which has been

completely or partially deoxidized by the addition of small portions of other metallic elements such as aluminum. Because rimming steel has a much lower carbon content it is inherently more ductile. However, the difficulty of maintaining a constant grain size as evidenced by the photomicrograph, *Fig.* 14, detracts somewhat from the deep-drawing property of this material. In this illustration the characteristic decrease in grain size toward the center of the specimen is strikingly evident. True, these small grains which have such a serious effect upon ductility could be increased in size by higher normalizing temperatures. However, in so doing there would be a strong likelihood of similarly increasing the size of the larger grains nearer the surface. As previously indicated, such an increase in grain size might induce various types of surface imperfections as well as tending toward premature failure in the drawing operation caused by necking down of the strained large grain size section and its inability to transmit the greater loads necessary to draw or deform the small grain sections.

Deoxidized with Aluminum

Killed steels ordinarily have a higher carbon content and hence are more difficult to draw, and large reductions per draw are not possible. However, these deoxidized steels have many important applications particularly where age-hardening is a critical factor.

Another photomicrograph of rimming steel is shown in *Fig.* 15. Here by proper heat treatment the grain size has been increased to increase the ductility of the metal. The dark spots in this illustration denote the presence of oxide and other nonmetallic inclusions. These have a deleterious effect on drawing properties and should be held to an absolute minimum.

Causes Discontinuity of Structure

Effects on grain structure and crystal orientation of various press-working operations are illustrated by *Figs.* 16 and 17. In the former, distortion of individual grains and the orientation of groups of grains in straight or continuous lines is particularly evident. *Fig.* 17 demonstrates a condition likely to occur as the result of a final finishing operation such as restriking, coining or embossing on a drawn part. Such an operation might result in a similar discontinuity of the grain structure, causing brittleness in the finished part. The condition illustrated by *Figs.* 16 and 17 can be relieved by a subsequent annealing operation.

Importance of avoiding directionality of grain structure in metal intended for deep drawing has been previously emphasized. In steel this property is more difficult to control than in the brasses. This is largely due to inevitable, elongated, nonmetallic inclusions as well as stringers of pearlite.

Machine Design asknowledges with appreciation the co-operation of the following companies in the preparation of this article: Aluminum Company of America, American Brass Co., and the Chase Brass and Copper Co.

Business and Sales Briefs

TIMKEN Roller Bearing Co., Canton, O., recently conducted a four-day campaign and sold $36,417.10 in Defense Savings Stamps. This is one of the pioneer efforts of its kind by a private corporation, and the action indicates how patriotism and ingenuity of an organization in co-operation with the U. S. Treasury can be turned into vital aid.

♦

Clark Controller Co., Cleveland, has appointed Charles H. Armstrong as assistant district manager of the Michigan sales territory.

♦

Another expansion program is being launched by Ampco Metals Inc., Milwaukee, Wis. This is in addition to its new bronze foundry recently dedicated, and the new machine shop.

♦

Formerly manager of motor sales division of Wagner Electric Corp., H. A. Hudson has been promoted to the position of manager of sales, electrical division.

♦

To meet increased demands, Reliance Electric & Engineering Co. has purchased the plant of Cleveland Hobbing Machine Co., the latter company moving its operations into a new plant at Euclid, O.

♦

An eastern sales office at 249 High street, Newark, N. J., has been opened by Keystone Carbon Co. Robert McKeown and Charles V. Allen are district representatives.

♦

J. M. Chapple has been appointed manager of a newly opened office of Lincoln Electric Co., at 700 East Union street, Jacksonville, Fla.

♦

Located at the Chicago office of Cutler-Hammer Inc. for the past 16 years, E. C. Bolton has been appointed manager of the Cincinnati district sales office. Mr. Bolton has been with the organization for 19 years.

♦

According to an announcement made by Charles E. Wilson, president of General Electric Co., five new vice presidents have been elected. They are: Walter R. G. Baker, Chester H. Lang, David C. Prince, Elmer D. Spicer, and Harry A. Winne.

♦

Previously connected with American Brake Shoe & Foundry Co., San Francisco, Damon Wack has been appointed assistant to the vice president in charge of sales, National Bearing Metals Corp., St. Louis, a subsidiary.

♦

To increase its capacity for rubber covering and rubber lining of tanks and machine parts of all types in large dimensions, American Wringer Co., Woonsocket, R. I., is installing a new vulcanizer in

its plant. The vulcanizer will take any object 33 feet long by 84 inches diameter, or 64 inches square.

♦

Removal of two sales offices and warehouses to new locations has been announced by The Bunting Brass & Bronze Co. The Atlanta, Ga., office has been moved to new quarters at 542 Spring street, northwest; and the Kansas City office to 1821 McGee street.

♦

At its plant in Arlington, N. J., E. I. du Pont de Nemours & Co. has opened a new plastics research center to provide for expanded investigations of plastics useful in national defense, and for the development of new materials.

♦

To be completed in about six months, a new windowless supercharger plant is under construction by Allis-Chalmers Mfg. Co., Milwaukee.

♦

For the manufacture of airplane exhaust manifolds and other aircraft parts, Solar Aircraft Co., San Diego, Calif., has purchased a plant at National City, Calif.

♦

To triple the production of diesel engines for the United States Navy, Fairbanks, Morse & Co. will build a new plant in Beloit, Wis.

♦

Formerly manager of sales of Lukenweld Inc., Coatesville, Penna., William S. Wilbraham has been promoted to manager of costs. Robert C. Sahlin, who has been assistant manager of sales, is now manager of sales.

♦

For the past year sales manager of the air conditioning and commercial refrigeration department at the Bloomfield, N. J. office of General Electric, Elliott Harrington has been named manager of sales of the Schenectady induction motor section of the General Electric motor division.

♦

Robert J. Howison, who has had more than 20 years' association with the silent and roller chain industry, has been appointed sales manager of the automotive division, Morse Chain Co., Detroit.

♦

Ohio Crankshaft Co., Cleveland, is celebrating its twenty-first anniversary. The company has four plants, one being completed recently for defense work, producing crankshafts for aviation motors. Production of the company's induction surface hardening units is up 300 per cent.

♦

Under the auspices of the Defense Plant Corp., Revere Copper & Brass Inc. is building a mill at Baltimore to manufacture condenser tubes for the government. Construction was commenced October 21 and the plant is scheduled for operation early in May, 1942.

♦

Wencel A. Neumann Jr. has been appointed to the office of president of Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa., succeeding George G. Main

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who since 1938 had been president of the company. Mr. Main is now assistant director, financial accounts division.

♦

The Defense Plant Corp., subsidiary of the Reconstruction Finance Corp., has authorized agreements for plant and machinery expansion to the following companies: Cleveland Graphite Bronze Co., Cleveland; Extruded Metals Defense Corp., Grand Rapids, Mich.; Vickers Inc., Detroit; Wellman Bronze & Aluminum Co., Cleveland; and Youngstown Sheet & Tube Co., Youngstown, O.

♦

In its small motor division at Lima, O., Westinghouse Electric & Mfg. Co. has started a plant-wide construction program. This is to meet increased defense needs for small electric motor generators. It is reported by the company that this is not the usual expansion program but mainly a rearrangement of facilities to increase manufacturing productivity.

♦

Working out of the New York office of Reynolds Metals Co., Richmond, Va., James R. Scully will act as general salesman for the company.

♦

Cooper Alloy Foundry Co., Elizabeth, N. J., will soon move to Hillside, N. J. It will occupy a new foundry to be in operation December 1.

♦

Succeeding L. W. Harston, resigned, W. J. Sampson Jr. has been appointed general manager of sales of the Steel & Tubes Division of Republic Steel Corp.

♦

Associated with General Electric Co. since 1922, Dr. H. A. Jones has been named manager of sales of the company's electronic tubes for nonradio applications in industry.

♦

Formerly assistant to the president of The Carpenter Steel Co., R. V. Mann has been named general sales manager. He has been with the Carpenter organization since 1911.

♦

A new factory and office building at Imperial Highway and Alameda street, Lynwood, Calif., will be built by Western Gear Works.

♦

Formerly assistant sales manager, Ben H. Jones has been made assistant vice president of National Screw & Mfg. Co., Cleveland.

♦

Opening of an Eastern sales office at 249 High street, Newark, N. J., is announced by The Keystone Carbon Co., St. Marys, Pa. Robert McKeown and Charles V Allen are district representatives. Another appointment, that of A. A. Barbera & Co., 417 South Hill street, Los Angeles, as representative in the Southern California territory, has been made by the company.

♦

Torq Electric Mfg. Co., Cleveland, O., has organized a rush job department to aid defense manufacturers to get equipment more quickly.

Meetings and Expositions

Jan. 6-8—

Institute of Scrap Iron and Steel Inc. Annual convention to be held at Sherman hotel, Chicago. Edwin C. Barringer, 1626 K street, N. W., Washington, is secretary.

Jan. 12-15—

Refrigeration Equipment Manufacturers association. All-industry refrigeration and air conditioning exhibition to be held at Stevens hotel, Chicago. Additional information may be obtained from Theodore R. Sills, 43 East Ohio street, Chicago.

Jan. 12-16—

Society of Automotive Engineers Inc. Thirty-seventh annual meeting to be held at Book-Cadillac hotel, Detroit. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary and general manager.

Jan. 15—

American Railway Car institute. Annual meeting to be held in New York. Additional information may be obtained from W. C. Tabbert, secretary, 19 Rector street, New York.

Jan. 21—

Aluminum association. Annual meeting to be held at 420 Lexington avenue, New York. Kenneth G. Castleman, at the foregoing address, is secretary of the association.

Jan. 26-28—

National Warm Air Heating and Air Conditioning association. Annual meeting to be held at Benjamin Franklin hotel, Philadelphia. George Boeddener, 145 Public Square, Cleveland, is managing director.

Jan. 26-29—

American Society of Heating and Ventilating Engineers. Annual meeting and exposition to be held at Bellevue-Stratford hotel, Philadelphia. A. V. Hutchinson, 51 Madison avenue, New York, is secretary.

Jan. 26-30—

American Institute of Electrical Engineers. Annual winter convention to be held at Engineering Societies building, New York. H. H. Henline, 33 West Thirty-ninth street, New York, is secretary.

Feb. 5-12—

National Electrical Manufacturers association. Annual meeting to be held at Palmer House, Chicago. R. J. Blais, 155 East Forty-fourth street, New York, is convention manager.

Feb. 23-28—

Motor and Equipment Wholesalers Association. Annual meeting and exhibit to be held at Atlantic City. C. W. Hammond, 1125 Columbia street, San Diego, Calif., is secretary.

March 4—

American Society for Testing Materials. Spring meeting to be held in Cleveland. R. E. Hess, 260 South Broad street, Philadelphia, is secretary.

March 10-13—

American Society of Bakery Engineers. Annual meeting to be held at Edgewater Beach hotel, Chicago. Victor E. Marx, 1541 Birchwood avenue, Chicago, is secretary.

POSITIONS

AVAILABLE OR WANTED

AVAILABLE: Thirty-nine year old American born mechanical engineer available immediately for either domestic or foreign service. Eleven years experience designing special machinery. Eight years in oil field, one in mining and two in aircraft industries. Have worked in both Central and South America. Capable of taking charge of department. Minimum salary $500 per month. Address Box 150, MACHINE DESIGN, Penton Bldg., Cleveland, Ohio.

WANTED: Machine designer. Experienced designer—mechanical engineer preferred—to expand and develop improved lines of equipment of the following types—mechanical presses, plastics molding equipment, high vacuum pumps, vacuum processing apparatus, as well as certain types of automatic machinery. An excellent opportunity for an outstanding creative designing engineer, with proven ability. F. J. Stokes Machine Company, Olney Post Office, Philadelphia, Pa.

PROGRESSIVE COMPANY: With Modern Plant manufacturing medium and heavy machinery, having highest financial rating, has retained us to investigate and recommend to them propositions of merit for manufacture as soon as existing demands through defense priorities make this possible. Particularly interested in new processes or unusual machinery or products having exceptional marketing possibilities. Assistance such as research, engineering, etc., might be furnished if essential. Propositions submitted will be considered confidential. Barkley Associates, 131 Clarendon St., Boston, Mass.

WANTED: Mechanical engineer. An unusual opportunity for an energetic and creative young engineer, between ages of 24-35, to work on bimetal products. After sufficient training this position will lead to that of sales representative for a progressive company which is between Providence and Boston. Please give complete information covering age, education, experience, salary, etc. Address Box 152, MACHINE DESIGN, Penton Bldg., Cleveland, Ohio.

WANTED: Production engineer: starting salary $6,000 per year. Should be a graduate of an accredited mechanical engineering school. 30 to 35 years old, to be in charge of all engineering and planning activities for an old, well-established, financially sound company in the Chicago area, making a nationally distributed, high precision product, a substantial percentage of which is defense work.
Applicant should have had mass production experience, preferably on precision work; should have a combination of executive and engineering ability. Address Box 153, MACHINE DESIGN, Penton Bldg., Cleveland, Ohio.

WANTED: We want an executive engineer; an engineer who has had a well rounded and varied experience in design and practical engineering practice, who is competent to direct and supervise the design and construction of modern industrial furnaces. He will be associated with one of the largest manufacturers in the field, and if he can qualify, his position will be permanent. Salary will be commensurate. Give full details of schooling, experience, age etc. with photo in first letter. Our own employees know about this advertisement. Address Box 154, MACHINE DESIGN, Penton Bldg., Cleveland, Ohio.

NEW MACHINES—
And the Companies Behind Them

(For illustrations of United States war machinery see pages 72-73)

Air Conditioning
Liquid cooling systems, Worthington Pump & Machinery Corp., Harrison, N. J.

Bottling
Soaker-type automatic bottle washer, Girton Mfg. Co., Millville, Pa.
Glass-lined pasteurizer, Pfaudler Co., Rochester, N. Y.

Dairy
Electric churn, Alabama Mfg. Co., Birmingham, Ala.

Domestic
Portable electric beater, Knapp Monarch Co., St. Louis.
Coal stoker, Conco Corp., Mendota, Ill.

Food
Solvent vaporizer, F. J. Stokes Machine Co., Philadelphia.
Batch scale car, Cleveland Tramrail Div., The Cleveland Crane & Engineering Co., Wickliffe, O.
Butter patty forming machine, C. Doering & Son, Chicago.
Barbecue machine, Hammond Machine & Tool, Hammond, Ind.
Ice cream freezer, Taylor DuMore, Beloit, Wis.

Heat Treating
Controlled atmosphere furnace, The Sentry Co., Foxboro, Mass.
Heat treating furnace, R-S Products Corp., Philadelphia.

Industrial
Semi-automatic drum opener, Turner & Seymour Mfg. Co., Torrington, Conn.

Laboratory
Plus and minus scale, The De Raef Corp., Kansas City, Kans.
Heavy-type balancing machine, The Globe Tool & Engineering Co., Dayton, O.
Spectograph, Harry W. Dietert Co., Detroit.

Metalworking
Turret lathe, Merritt Engineering & Sales Co. Inc., Lockport, N. Y.
Propeller blade milling machine, Sundstrand Machine Co., Rockford, Ill.
Borer and driller, Merz Engineering Co., Indianapolis.
Drilling machine, Avey Drilling Machine Co., Cincinnati.
Belt sander-grinder, Porter Cable Machine Co., Syracuse, N. Y.
High-speed precision milling machine, Jefferson Machine Tool Co., Cincinnati.
Down-cut milling machine, Bremacc Corp., Detroit.
Milling machine, C. C. Bradley & Son Inc., Syracuse, N. Y.
Spot facing machine, Taft Pierce Mfg. Co., Woonsocket, R. I.
Hydraulic hack saw, L. B. Mfg. Co., Los Angeles.
Power feed drill press, Walker-Turner Co. Inc., Plainfield, N. J.
Milling machine for aircraft engines, Moline Tool Co., Moline, Ill.
Boring mill, Ohio Machine Tool Co., Kenton, O.
Power squaring shears, Niagara Machine & Tool Works, Buffalo.
Straightening press, Denison Engineering Co., Columbus, O.
Hydraulic press, Bawden Bros. Inc., Davenport, Ia.

Quarry
Air-cooled ultraviolet lamp, Hanovia Chemical & Mfg. Co., Newark, N. J.

Surgical
Crusher, American Pulverizer Co., St. Louis.
Roll crusher, Diamond Iron Works Inc., Minneapolis.

Welding
Cold arc welder, A C Devices Co., Chicago.
Spot welding machine, Pier Equipment Mfg. Co., Benton Harbor, Mich.
Welding torch, Alexander Milburn Co., Baltimore.
Forge-welder, Progressive Welder Co., Detroit.